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[621 .337]

Individual axle drive.

Mechanical systems used on electric locomotives and motor coaches, with an indication of the results obtained in service on railways of all kinds,

by Adolphe-M. HUG,
Consulting Engineer, Thalwil (Zurich), Switzerland.

Note. — This article constitutes an up-to-date revision (Autumn 1952) of the work by the same author, the FIRST VOLUME of which appeared in **1933** (reprint of a revised and amplified study which was published in *La Traction électrique*, Paris, 1931-1932) and the SECOND VOLUME in **1950** (reprint of revised and completed articles published from 1946 to 1949 in 13 numbers of the *International Railway Congress Bulletin* (French edition), Brussels) (*).

The *first* study of this subject was undertaken by the author on the occasion of the International Railway Congress in Madrid in 1930, and was published in the *Congress Bulletin* of June 1930 (English edition), of March 1930 (French edition), and of September 1930 (German edition).

EXPLANATORY NOTE.

The author has dealt with the various types of flexible transmission in the same order (and distribution) as in 1947/1950 (mentioned in the note above) and in the Table of Contents, page 5 of Volume II of 1950 (*).

Resilient wheels, dealt with in Chap-

(*) For the English edition, the issues of the *International Railway Congress Bulletin* were :

1947 Numbers for September, October and December (Chapters I, II, III and IV);
1948 Numbers for February, April, July, October, November (Chapters IV [*continued*], V, and VI);
1949 Numbers for January (Chapter VII), May and November (Supplement. Appendix);
1950 Number for January (Appendix [*continuation and end*]).

The two volumes of the complete work, published by Birkhäuser, Basle (Switzerland);
Volume I, 1933. — 112 pp., 202 figs, with French/German/English technical glossary;
Volume II, 1950. — 320 pp., 397 figs, with index by countries and classes (comprising all the articles mentioned above).

Note. — In the present study, these two issues are designated as *Vol. I* and *Vol. II.*

ters IV and VI (IV, V and VI of Vol. II), with the relative mechanism of motor coaches, have on this occasion been grouped in Chapter II, since they are particularly concerned with « nose-suspended » traction motors.

As in previous works, we shall not repeat what has already been written (in the 13 issues indicated and in Volumes I and II), but we shall describe, as far as possible with an *assessment and note of behaviour in service*, new arrangements and fresh applications which have occurred in the meantime — from *1949 to 1952* inclusive.

The numbering of the illustrations and notes at the foot of a page (particularly bibliographical) is continued, practically, from the January 1950 issue of the *Congress Bulletin* (pp. 17-46) (*).

With the aim of simplifying the bibliographical notes, we shall adopt in this new supplement, the following abbreviations of the titles of technical publications:

- *International Railway Congress Association Bulletin*, Brussels : *Congress Bulletin;*
- *International Railway Congress Association Bulletin — Electric Traction on the Railways*, Brussels : *Congress Bulletin T.E.;*
- *Revue Générale des Chemins de fer*, Paris : *R.G.C.F.*;
- *Revue de l'Association française des Amis des Chemins de fer*, Paris : *A.F.A.C.*;
- *Elektrische Bahnen*, Munich : *E.B.*;
- *Verkehr und Technick*, Berlin-Bielefeld-Munich : *V. & T.*;
- *Railway Gazette*, London : *Gazette*;
- *Economie et Technique des Transports*, Zurich : *E.T.T.*

The author particularly wishes to call the attention of readers and others interested to the fact that the field of axle drive mechanisms has, during the past five years, in numerous countries, been subject to such expansion that the present article cannot pretend to be nearly complete (**), particularly in regard to the very extensive bibliography. The author consequently asks to be excused for any shortcomings in the present article. In addition, it may be said that space is insufficient for the inclusion of *everything* which it would be of interest to include.

CHAPTER I.

GENERAL.

(and rod coupled locomotives) (241).

Apart from three phase locomotives (see figs. 3 to 5 of Vol. I) (242), the

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(*) The numbering of the figures in the supplement thus commences at Figure 398; Volume II contains nine additional illustrations, inserted from page 59 with duplicate (i.e. *a*, *b*) numbers, during compilation.

(**) See Preface to Volume II, by Mr. L. ARMAND, General Manager of the SNCF (French National Railways), dated 18th April 1950.

(241) See *Congress Bulletin*, Nos. for September 1947, pp. 825-830 (figs 1-7) and May 1949, pp. 379-382 (figs. 276-278).

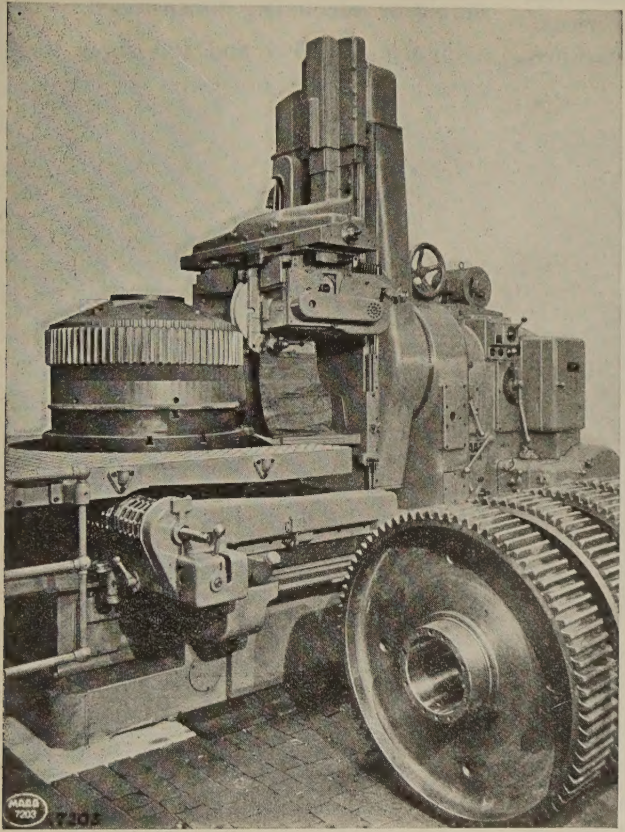
Vol. I (pp. 3-7, figs. 1-7) and Vol. II (pp. 4-10, figs. 1-7 and 276-278).

(242) On 3-phase main line locomotives, rod control was almost inevitable since, at the beginning of the century, the power of a 3-phase motor was too high for one axle and had to drive two or three, without gears. Now, individual flexible transmission from motor to axle would be possible, but 3-phase traction is gradually being abandoned for various well-known reasons (double contact line, shunt characteristics of motors, rigidity of speed, etc.).

Swedish State Railways S.J. (Statens Järnväger) and shunting tractors (in many cases and various countries) *rods* are used in the drive less and less, other than for the coupling of axles, but the general tendency (even with hydraulic transmission of Diesel engined machines) is more and more for *individual drive* for each driving axle.

It may be noted that the Swedish State Railways S.J. are to put into service two new series of *rod-coupled locomotives*, for express and local trains, as follows :

1) in 1953, for the mountain line Luleå-Narvik (in the north) 1'D + D1' locomotives for a maximum speed of 75 km (47 miles)/h with an hourly rating at the armature shaft of 2 × 2 400 HP (at a speed of 52 km [32 miles]/h); the load per driving axle is 17 t and provision is made for this to be increased later to 20 t;



Block from: « Economie et Technique des Transports » (E.T.T.).

Fig. 398.— Case-hardened, tempered and ground gear wheels of electric locomotives, figures 244 and 366, each transmitting an hourly power of 1 000 HP (per axle, compare with figures 246 and 247) of B_0-B_0 locomotives, Ae $^4/_4$, series 251, of the Bernese Alps (BLS) Railway; maximum speed in service : 125 km (78 miles)/h; axle load : 20 tons.

These wheels have 80 teeth, 170 mm wide, modulus 12.0 and are ground on a MAAG type SHS-120 machine.

2) a certain number of D class (« litt. Da ») locomotives, type 1′C1′ (see figs. 276-277, p. 380 of the *Congress Bulletin* for May 1949) (243) but with hourly rating increased to 2 400 HP and maximum speed 100 km (62 miles)/h, axle load 15 tons, which can later be increased to 17 tons if necessary; the provision of these machines commenced in 1952.

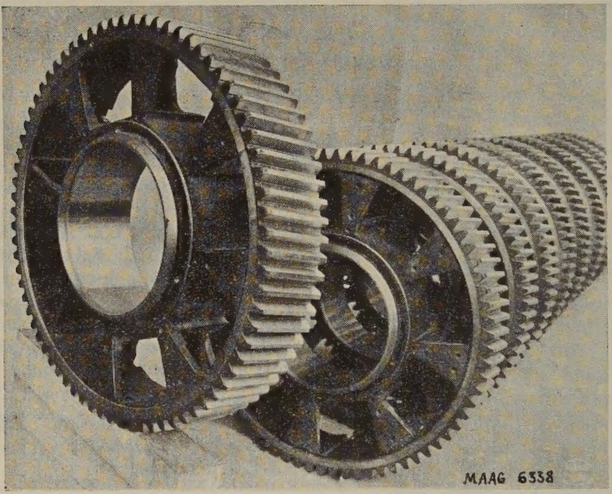


Fig. 399. — Case-hardened and tempered gear wheels ground on MAAG machine, of electric locomotives in figures 184-186, each transmitting an hourly rating of 575 HP (B_0-B_0 type locomotives, Re 4/4, series 401-456, for light express trains of the Swiss Federal Railways, CFF; maximum speed in service : 125 km/h; axle load : 14.2 tons).

Block E.T.T.

We shall deal briefly in the final Chapter VIII, with individual axle driving mechanisms for transmission of tractive effort on locomotives and motor coaches with *Diesel motors*, either mechanical or hydraulic (see, for example, *Congress Bulletin*, May 1949, fig. 291, p. 393) (244).

With regard to *gears*, it is most important, on account of maintenance costs and noise — we repeat this — that the rims should be case-hardened, tempered and ground. The most modern arrangements appear to give every satisfaction (see, for example, fig. 278, p. 382 of the May 1949 Bulletin) (245). It is also important that the areas of mesh and drive should be adequately dimensioned.

Figures 398, 399 and 400 show three typical examples of gears for modern locomotives and motor coaches in Switzerland. It is interesting to compare these three photographs with that of figure 278 mentioned above, and referring to the French National Railways, SNCF.

With regard to *axleboxes*, — a question which is almost as wide as that of driving mechanisms — railway engineers remain divided in opinion; according to operating conditions (and the development of the particular industry in the different countries), the new large classes of locomotives and motor coaches are ordered with roller bearings or with plain bearings and forced lubrication, this being the arrangement which, for motored stock at least, but in certain respects for trailer vehicles as well, seems preferable.

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(243) p. 218 of Vol. II (*Congress Bulletin*, May 1949, p. 380).

(244) In Vol. II, 1950, see fig. 291, p. 232; see also in Vol. II the final notes, p. 309, on this subject.

(245) p. 220 of Vol. II.

We have already given, in *Congress Bulletins* of May 1949 and November 1949 in figures 279-280 and 352 respectively (246), typical recent examples of each of the two fundamental principles, that is « Athermos » boxes (Isothermos, forced lubrication) and SKF roller bearing boxes which appear the most widely used, at least in Europe. These applications refer to locomotives and motor coaches of the main railway systems. In figures 401 and 402 are shown recent applications for large classes of plain bearing boxes with forced lubrication; some others will also be shown in the descriptions of certain locomotives.

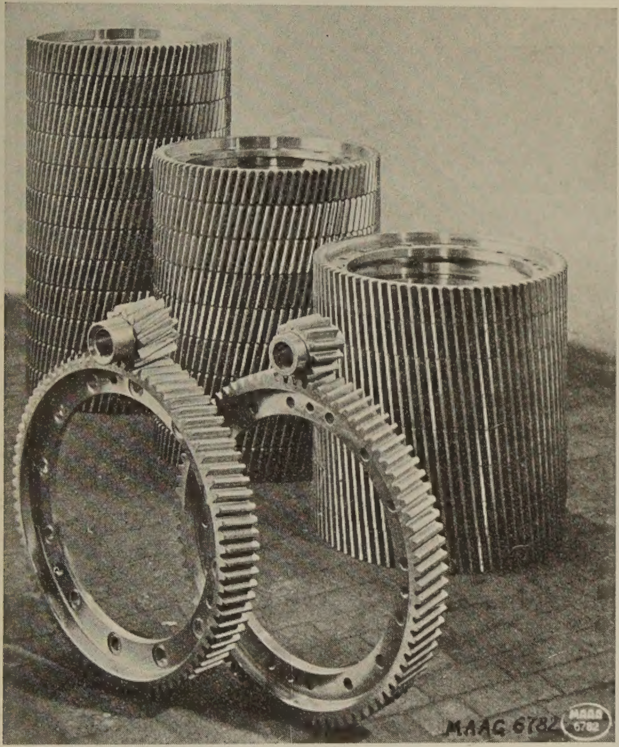

Fig. 400. — Case-hardened, tempered and ground gears for the recent high capacity motor coaches of the Swiss town tramways (Zurich, Basle, Geneva, Berne, Lucerne). Power transmitted per axle, 70-90 HP, at 1 530 r.p.m. (maximum speed : 2 900 r.p.m.).

(Compare with figures 107*a* and 107*b*, 250 to 253, 378 and 380.)

Note. — The Zurich Transport Co. are no more called St. St. Z., as indicated previously, but VBZ, « Verkehrsbetriebe der Stadt Zürich ». It is the largest operating system after the Swiss Federal Railways, CFF.

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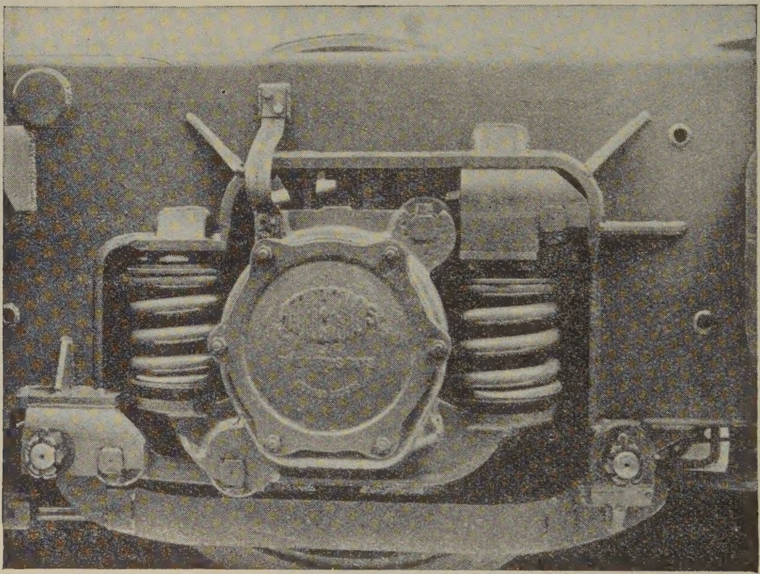
(246) p. 22 and p. 276, Vol. II respectively.

CHAPTER II.

MOTORS WITH « NOSE » (or tramway type) SUSPENSION AND RESILIENT WHEELS (247).

All the main points have already been mentioned in the September 1947 and May 1949 issues (pp. 831-836 and 383-385 respectively). There has been scarcely any change during the last three or four years, either in Europe or overseas, apart from numerous applications which have been made in the meantime. Mention will be made, later in this chapter, of some special or extensive applications.

In connection with transmission incor-



Isothermos Photograph.

Fig. 401. — Athermos axle box with special Alsthom suspension, with silentblocks of the locomotives figure 292 (B_0-B_0 electric locomotives; series 1101, Netherlands (NS) railways). These are CFN[1] boxes for 160 × 260 mm (5′ 3″ × 8′ 6 3/8″) journals, 500 being supplied in 1950/51 for these locomotives. (Compare with figure 434.)

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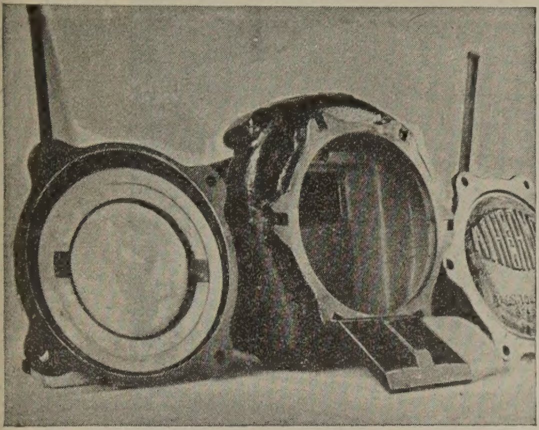
(247) See *Congress Bulletin*, September 1947, pp. 831-836 (figs. 8-12) and May 1949, pp. 383-385 (figs. 279-280).

Also in Vol. I, pp. 11-21 (figs. 16-37) and in Vol. II, pp. 11-16 (figs. 8-12) and pp. 221-224 (figs. 279-280).

With regard to *flexible wheels*, see Chapters IV and VI in the *Congress Bulletins* for November 1948, p. 692 (fig. 255) and November 1949, pp. 785-786 and 801-802 (figs. 319 and resp. 341), and January 1950, pp. 29-31 (figs. 369-371).

In Vol. II, p. 184 (fig. 238*a*), p. 199 (fig. 255), pp. 252-253 (fig. 319), pp. 268-269 (fig. 341) and pp. 291-292 (figs. 369-371).

porating rubber pads, mention was made of trials being carried out by the French National Railways, SNCF, since 1951 with *resilient wheels* of the SAB and SAGA types [see (247) resilient wheels].



Isothermos Photograph.

Fig. 402. — Athermos axlebox dismantled, type CFEF[1], for tender bogies of the Finnish (VR) State Railways, 1951.

In respect to these trials on four BB locomotives of the SNCF, we may point out that this Company has put into service successively since 1947, a fairly large number of BB locomotives with nose-suspended motors. Apart from the prototype No. 8001 (formerly numbered 0401) supplied at the beginning of 1947, 136 locomotives (Nos. 8101 to 8236) are now in service, the earliest of these having been in service since the end of 1948. Thirty-five others are at present on order (1952, Nos. 8237 to 8271) (248). The gear wheels also have flexible rims (see figs. 403 and 405, gear at one side of axle only).

With regard to the trials in progress on four of the new BB locomotives, with

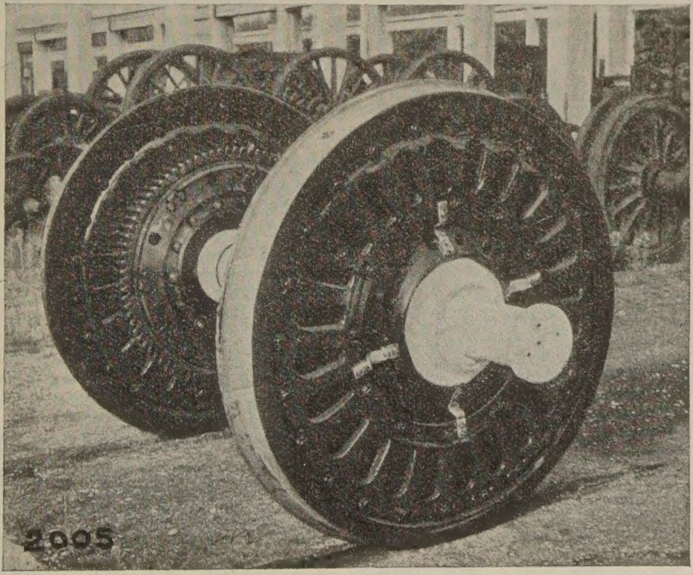


Fig. 403. — Complete driving axle of a BB locomotive (SNCF, No. 8189, 1951) fitted with flexible wheels, Paulstra-SAGA-Pirelli type (see text). Dia. of driving wheels (on tread) : 1 400 mm (4′ 7 1/8″).

The gear wheel, clearly visible, has a flexible rim with coil springs, mounted against the periphery in Z form, as can be seen at the right of the wheel centre on the left of the illustration in figure 341.

SNCF-4 Photograph.

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(248) Engines prior to 1947 were largely in accordance with figure 9 (p. 832 of *Congress Bulletin*, September 1947, = p. 12 of Vol. II).

resilient wheels, the SNCF has given the following information :

The Swedish SAB (Svenska Aktiebolaget Bromsregulator, Malmö) device, as shown in figures 238*a*, 404 and 405 has been used for all wheels of two BB locomotives of the 8101 class (8109 with 20 tons axle load and 8233 with 23 tons axle load). Since these SAB wheels were fitted, the two locomotives had run, up to the end of September 1952, 218 000 and 152 000 km (135 500 and 94 450 miles) respectively, without defect or difficulty. The trials continue.

With regard to the Paulstra arrangement [SAGA-Pirelli licence of Italian origin, see fig. 341, p. 801 of the *Congress Bulletin* for November 1949 (p. 268 of Vol. II) (247)], the first set of 8 wheels fitted to BB locomotive, No. 8161, ran 25 000 km (15 540 miles) before being dismantled temporarily for adjustment (fig. 341). They are to be refitted on locomotive No. 8189, the 75th with

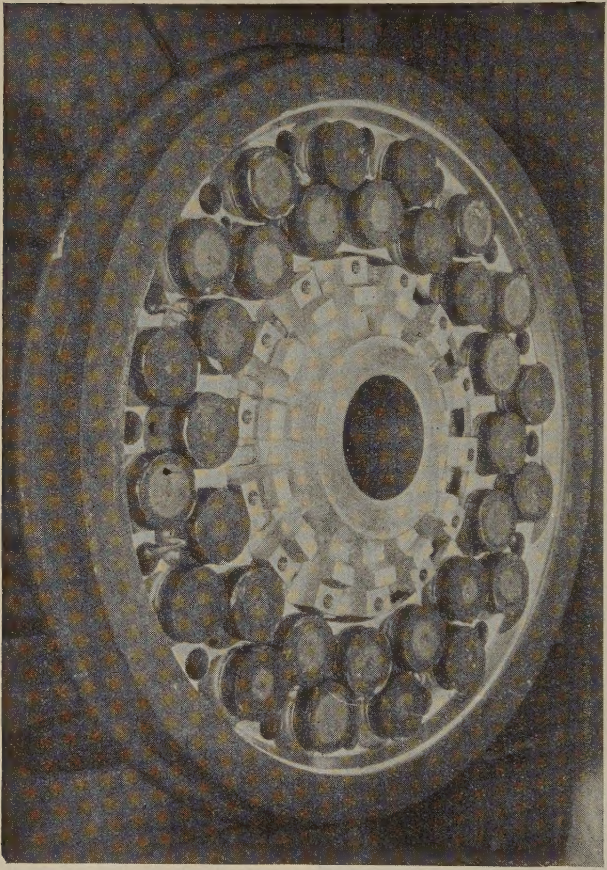
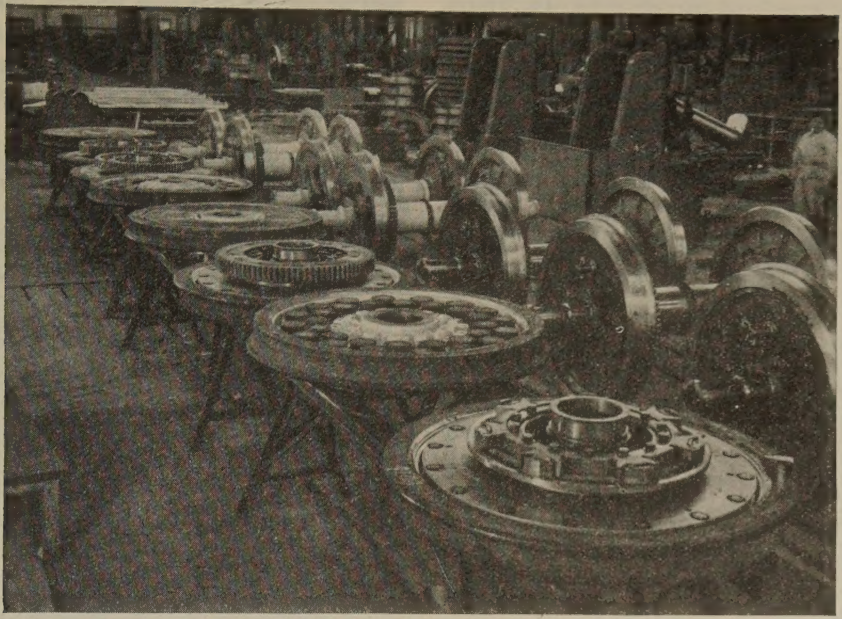


Fig. 404. — View of an SAB wheel, type 193, for locomotives, with outer disc removed, showing the arrangement of the 32 rubber discs and the fixing of the outer disc. This view corresponds exactly with the plan in figure 238*a* (p. 184 of Vol. II).

(SNCF locomotive, B_0-B_0 type, No. 8109.)

Photograph: Société Alsacienne, Graffenstaden (January 1951).



Photograph : Société Alsacienne, Graffenstaden (January 1951).

Fig. 405. — View of a set of SAB wheels in various stages of assembly, showing the flexible geared rim and its fixing (cf. figs. 238*a* and 404). In the background a set of axles with rigid wheels and Alsthom drive mechanism with hollow shaft and silent blocks for locomotives 7001 and 7002. (Locomotives SNCF.)

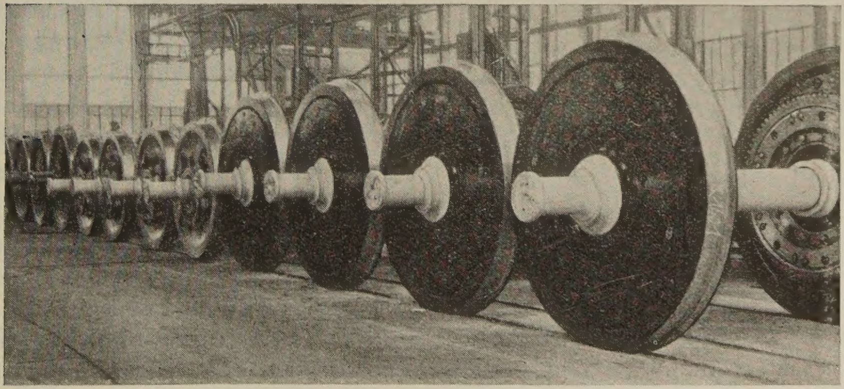


Fig. 406. — Sets of axles for BB electric locomotives. The first four have SAB wheels (figs. 404 and 405 SNCF); the next four have Alsthom mechanism (Netherlands State Railways, NS) and the last four have standard BB locomotive wheels, SNCF.

23 tons axle load, supplied by Als-Thom. The second set are also to be used on a BB locomotive of the same series.

Figure 403 shows one of the assembled axles of the Paulstra-SAGA set recently installed on locomotive 8189. Around the centre can be seen the 4 copper bonding cables for return current and the rubber segments which insulate the rim from the boss.

Figure 404 shows one of the SAB wheels of locomotives 8109 and 8233 with the outer disc removed; the arrangement of the 32 small rubber discs can readily be seen. Figures 405 and 406 show in the foreground a set of SAB wheels and in the front can be seen the body (boss portion) of the geared rim, flexibly mounted on springs as already described.

For these resilient wheels it is preferable to use *synthetic* rubber which is unaffected by oil and grease.

In general, the SNCF (which also intends to equip some motor coach sets with resilient wheels), is of the opinion that the trial appears conclusive, resilient wheels having given the engines great smoothness. The initiative shown by the SNCF in this respect is noteworthy.

The Swiss Federal Railways, SBB — CFF — FFS, have had in service since 1948 two *coaches* with SAB wheels, on the metre gauge (adhesion and rack) line of Brünig (Lucerne-Meiringen-Interlaken). The normal adhesion speed is 70 km (44 miles)/h and trials have been carried out at 75 km (47 miles)/h. The result has been that noise in running is rather less and above all that tyre wear is much less. These wheels were supplied by Aciéries G. Fischer + GF +, Schaffhouse, the Swiss agents for SAB.

It is no longer necessary, as is the case with coaches having standard axles, to send the wheels to works because of tyre wear. The CFF consider this to be a decided advantage and intend to fit SAB wheels to a large number of coaches. The reduced tyre wear is due to the fact that the rubber discs allow a slight transverse movement of the rim in relation to the centre.

We may also note that the Swedish State Railways, S.J., already mentioned, have recently (1952) ordered SAB resilient wheels for 280 four axled Diesel railcars; other applications for electric traction are also proposed.

The fitting of resilient wheels is often intended (and this is particularly the case with the SNCF) to counter the bad effects of the unsprung weight of nose-suspended motors, with which in effect half the weight of the motor/gears assembly rests without springs on the axle shaft. In addition, there are the difficulties occasioned by skidding (249) the hammer blows being countered owing to the action of the resilient wheels which absorb slack and vibration.

It seems premature to make any definite claim regarding the fitting of flexible wheels for heavy traction, but the trials by the SNCF already mentioned appear very encouraging. For motive units of tramway type [or motor coaches on secondary lines (250)] or for coaches inlocal service, experience is sufficient to recommend the fitting of resilient wheels, provided that there are no long gradients with continuous braking by shoes on the wheel rims (251).

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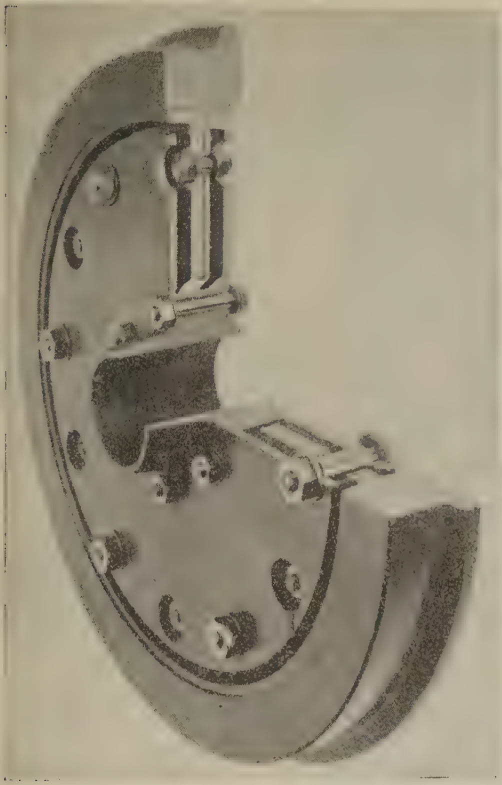
(249) See p. 14, Vol. I (fig. 21).



« Osnabrück » Photograph.

Fig. 407. — Exterior view and section of the Phönix flexible wheel, the type used on the Hamburg HHA tramways (about 800 wheels in service since 1950/51).

The most recent applications which have given good results over some years with modern motor coaches for tramways or similar lines are those shown in figures 107*a*, 107*b*, 251, 255, 371, 376, 378, and 380, the three inserts at the top of figure 409 and finally figure 414. The type of flexible wheel with a large number of small discs, shown in figure 370, has on the contrary been abandoned as it has not given very good results in service and is moreover too complicated in assembly and maintenance (252). The same applies to wheels as in figure 319. The fitting of these wheels entails careful attention to the remainder of the suspension; if there are too many rubber units, there may arise vibratory resonance which will rapidly destroy certain of the rubber elements and the assembly will not maintain sufficient rigidity (252).

Apart from the above mentioned two types which have been used on a large scale (those of which the principle is shown in figures 255 and 371) and which are of European design (Sweden, Italy), mention may be made of the American type PCC (of the Presidents Conference Committee [P.C.C.] of the American Transit Association, New York City) (see fig. 409) fitted over some fifteen years and with only few alterations (253).

A type of flexible wheel (of Spanish

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(250) See *E.T.T.*, Nos. 4-6, 1952 (Nos. 7-9 French edition), figs. 1, 4 and 6 in « Modernisierung der St. Gallen-Speicher-Trogen-Bahn », F. Joss.

(251) See *V. & T.*, May 1949, pp. 85-88, « Moderne Fahrzeuge für Lokalbahnen », seven figures, Ad.-M. Hug.

(252) See later remarks with regard to fig. 319.

(253) See *Verkehrstechnik*, Berlin, 20 May 1940, pp. 176-177, figs. 6 and 7, « Die neueste Entwicklung der Nahverkehrbetriebe im Nordamerika ». Ad.-M. Hug.

design) fairly similar to that in figure 371 has been developed and patented; it has been used since 1948 on the Barcelona City Transport system « TB » (Tranvias de Barcelona S. A.). This type also has fulfilled expectations.

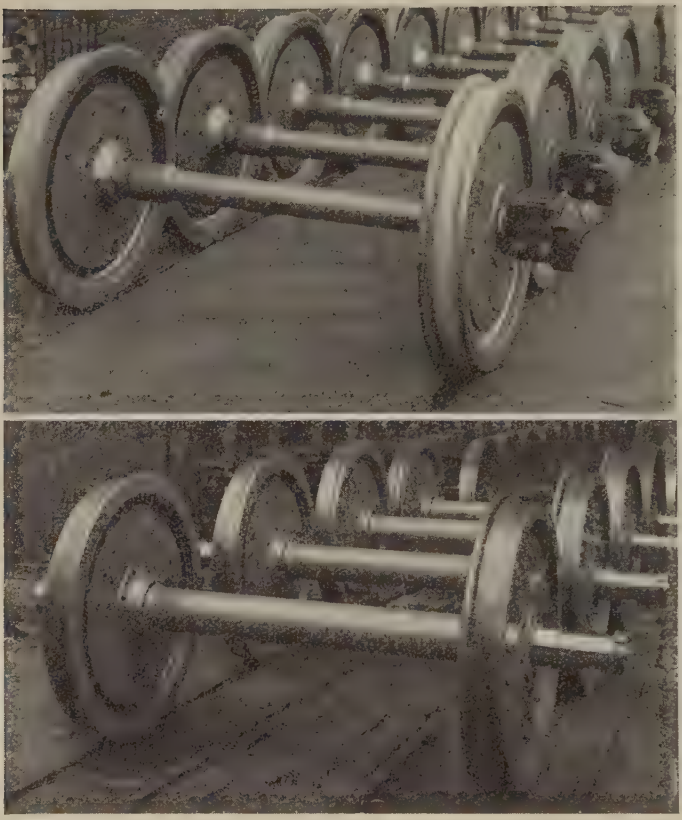

Fig. 408. — Tramway axles fitted with Bochum type flexible wheels (two somewhat different types).

Bochumer Verein Photograph.

In Western Germany, three types of flexible wheels have been developed since 1950, viz :

1) A type called « Phönix » from the Osnabrück Steel Co. (fig. 407) developed in association with Hamburger Hochbahn AG, H.H.A. (Hamburg City Transport — apart the « underground system » of the German railways, D. B.) and used by them (254).

2) A type from the « Bochumer Verein », Bochum (fig. 408), fitted on the new

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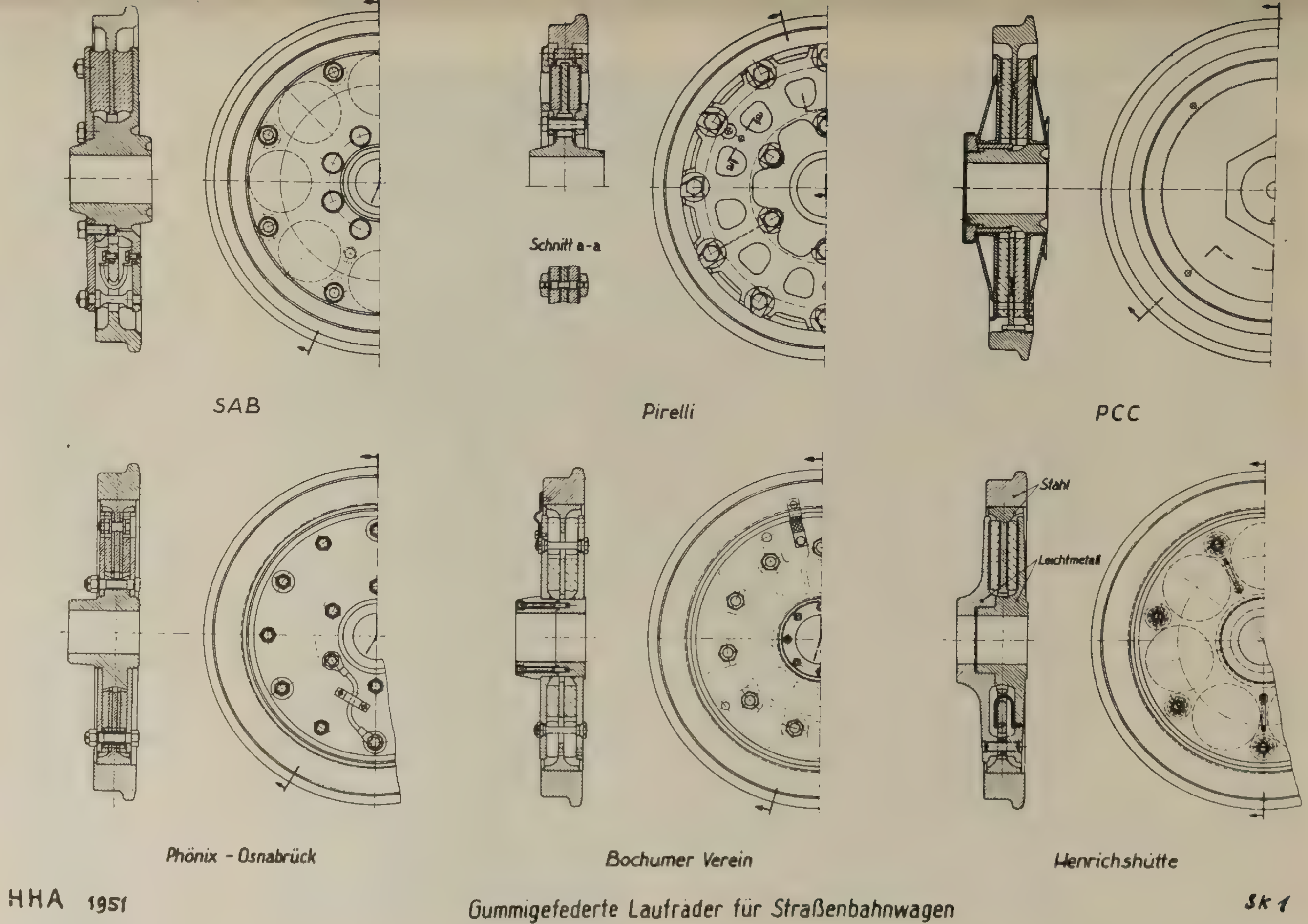
(254) See *V. & T.*, No. 2, 1952 : « Der neue PCC-Strassenbahn-Triebwagen der Hamburger Hochbahn AG », W. Lüttich; also *Aluminium*, Düsseldorf, No. 1, 1951, « Leichmetallfahrzeuge der Hamburger Hochbahn AG » (figs. 6 and 7), W. Lüttich.



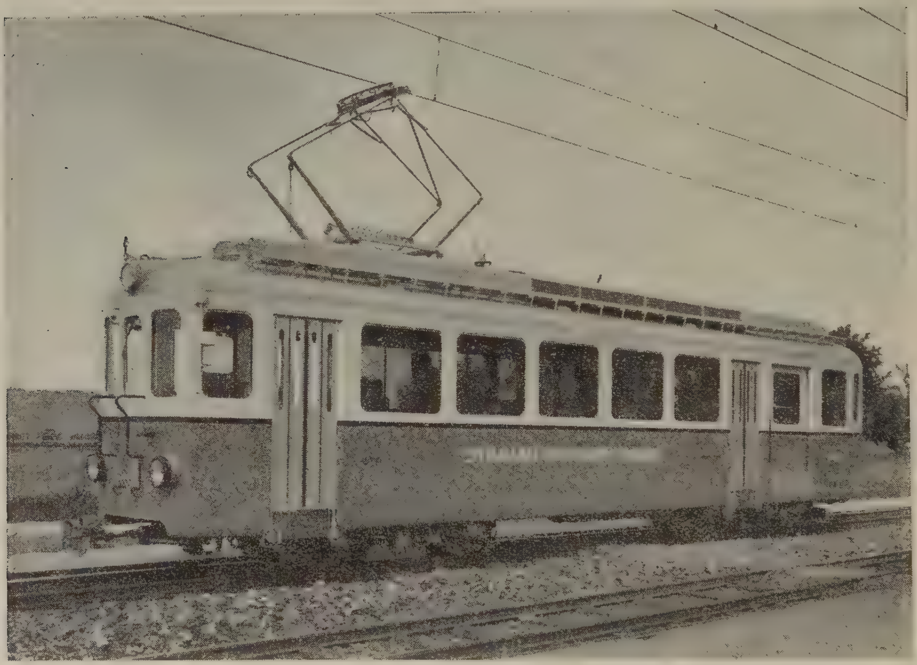
Plan H.H.A.

Fig. 409. — Comparative plan of the six most widely used flexible wheels (in section and elevation). Cf. the SAB in figures 255 and 238*a*, the SAGA-Pirelli in figures 371 and 341, the « Phönix » in figure 406 and the Bochum in figure 408.

motor coaches of the Hannover Transport « Ueberlandwerke und Strassenbahnen Hannover AG » ([255]).

3) Finally, that from the « Heinrichshütte Works» (Ruhrstahl), at Hattlingen (see fig. 409).

Nothing can yet be said about these three types of resilient wheels, their application being of too recent a date. The Hamburg HHA has however all three types on trial (4 with the PCC), but mainly the « Phönix » (92 four axles motor coaches); the « Bochum » type wheels (see fig. 408) already have more than 130 000 km (80 000 miles) in service ([256]).



Schindler Photograph.

Fig. 410. — Adhesion motor coach of the Trogen TB Railways (St. Gall-Appenzell cantons) with « tramway » suspended motors and flexible wheels.

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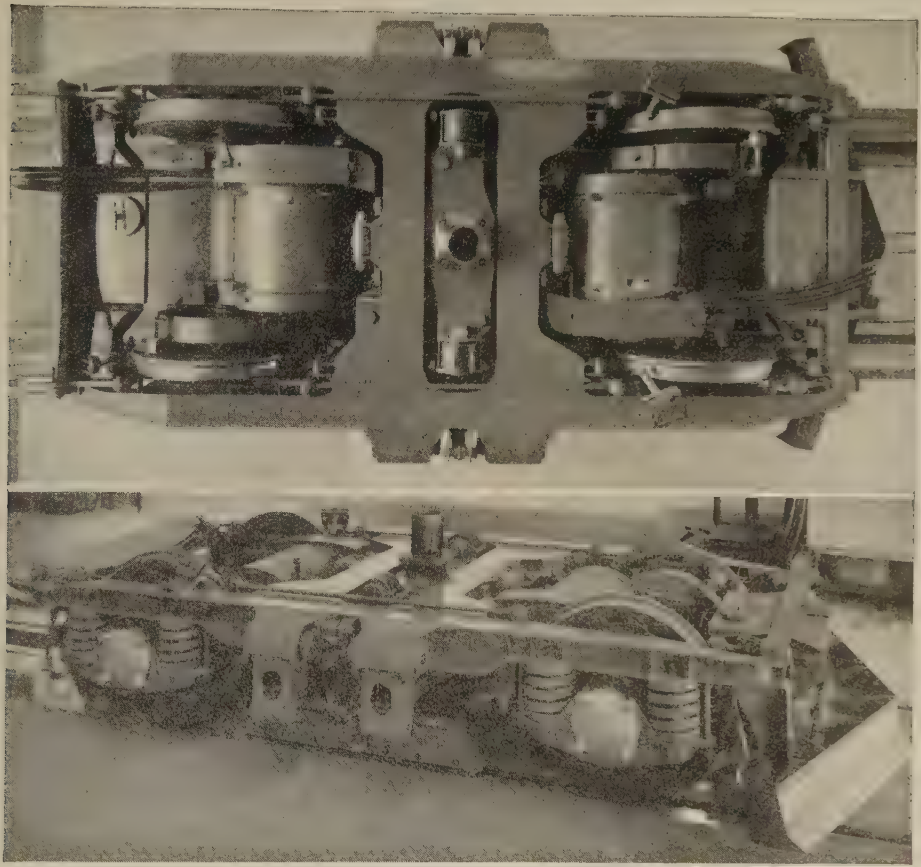
[255] See *V. & T.*, July 1951, p. 187, fig. 6 : « Strassenbahn-Grossraumwagen, Typ Hannover », Ph. KREMER.

[256] A fourth type « Conti » (abbr. for « Continental ») has been developed by the Uerdingen Wagon Co., but it does not seem to have any permanent application; it was the first in Western Germany and experience after the second world war appears to have been insufficient. — With regard to earlier types of resilient wheels in Germany, see *E.T.T.* (formerly « L'Allègement dans les transports ») Nos. 7-8, 1937, pp. 87-88 (figs. 3-5, Essen tramways S.E.G.), and « Roues élastiques dans la construction moderne de tramways », by A. v. LENGERKE, pp. 97-99 (figs. 1-2). Also by the same author : « Federnde Räder », *Verkehrstechnik*, Berlin, February 5, 1937.

The HHA, Hamburg Transport, mentioned earlier, have compiled a comparative statement of the six best types of flexible wheels. This very clear table is reproduced as figure 409.

We mentioned in note (^{250}) the three new electric motor coaches, Nos. 6, 7 and 8 of the St. Gall-Speicher-Trogen « Trogener Bahn » TB, Switzerland, metre gauge, adhesion line, with long gradients of between 55 and 73 $^{0}/_{00}$, the length of line is 9.2 km (5.65 miles) with a peak height of nearly 1 000 m (3 280 feet) above sea level. This is the first case in the world, so far as we know, where motor coaches for a local line with long gradients have been fitted with resilient wheels: the axle load (with the motor coach loaded)



Schindler Photograph.

Fig. 411. — Bogie of motor coach in figure 410, top and three quarter view. The flexible wheels, unilateral gear case, pivot and suspension components can be seen.

can go to about 9 tons and the brakes when descending are electrically operated. These motor coaches (fig. 410) have two bogies (fig. 411) and two motors to each, with nose-suspension (without flexible transmission mechanism), the hourly rating of each motor being 130 HP (at 1 000 V and a speed of 33.5 km [21.7 miles]/h. Figures 412 and 413 show, respectively, the flexible wheel, in section through the axle centre, and the axle suspension. In this case, therefore, — and this is why it is mentioned in some detail — the flexible wheels provide suspension not only for the weight of the assembled axle, but also that of the motors and gears resting directly on the axle.

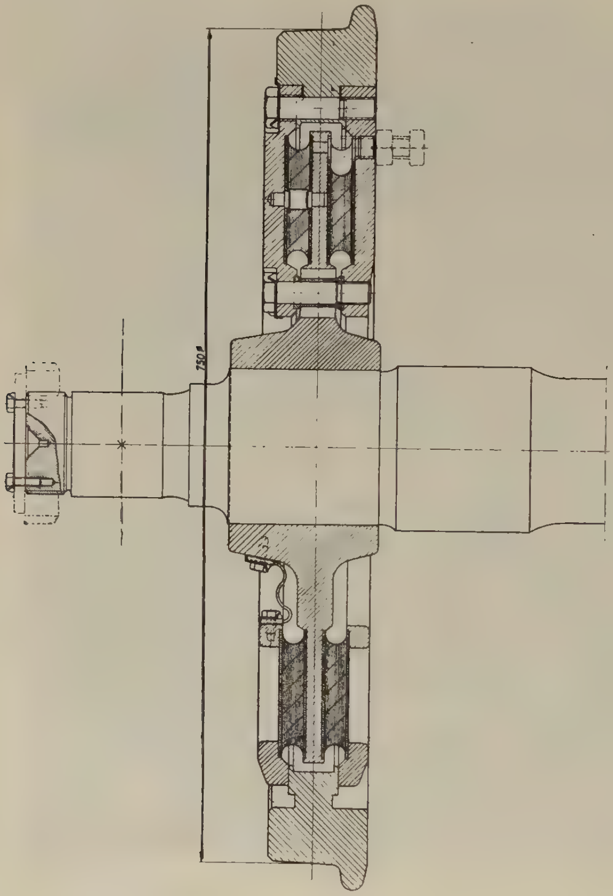
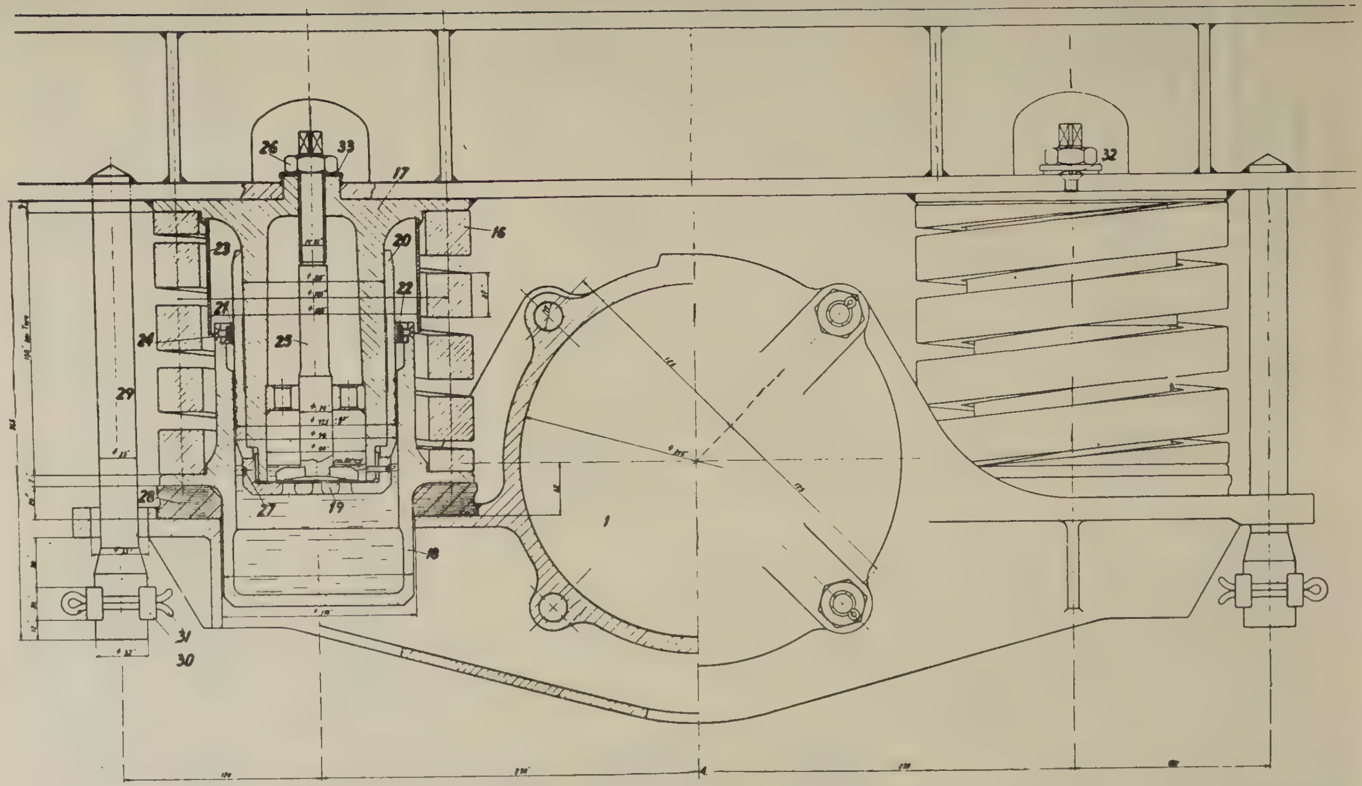


Fig. 412. — Flexible wheel with 120° rubber segments, Schindler-Pirelli type (SAGA) of motor coach in figures 410 and 411 (cf. with the second item of figure 409).

Plan Schindler

Still on the subject of flexible wheels (for heavy traction but with regard to a flexible transmission) we may say that the application in figure 319, already mentioned in this Chapter, has not been satisfactory and the rubber discs

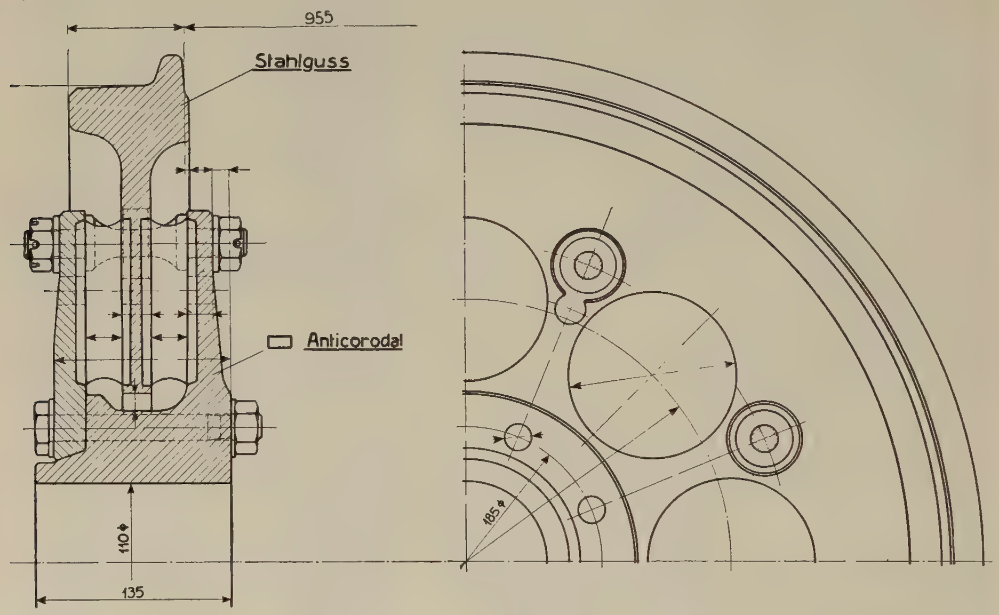


Plan Schindler.

Fig. 413. — Suspension and guide for axles of the motor coach in figures 410 and 411.

1 — axle box.
16 — coil spring.
17 — piston and top support for spring 16.
18 — cylindrical sleeve, bottom support for spring 16.
19 — base of piston 17, whith oil grooves.
20 — control box.
21 — stop carrier.
22 — felt washer.
23 — protecting sleeve.
24 — tightening screw.
25 — adjusting rod.
26 — hexagonal screw.
27 — spring washer.
28 — rubber washer.
29 — safety bolt.
30 — packing piece.
31 — pin.
32 — plug.
33 — zinc disc.

have deteriorated rapidly. The cause may be attributed to various factors; it is possible that rubber segments covering a quarter or a third of the periphery of the wheel discs would be preferable to *numerous* small discs; experience will settle this point in time. In addition, running speeds in this particular case are high (in the region of 130 to 150 km [80 to 93 miles]/h) with a relatively small wheel diameter (approx. 1 m [3′3 3/8]); finally, it may be — and this is probably the case here — that the vibration waves of the various components of the suspension (particularly if the latter is, even partially, also of rubber) and the transmission are superimposed and thus create an unstable condition in which the rubber components cannot long subsist.



Plan + GF +.

Fig. 414. — Section and elevation of SAB wheel, new tramway type, with boss and lateral discs, in light « anticorodal » alloy, either forged or cast (bogie trailers, 717 series, Zurich Transport system. VBZ, 1 m [3′ 3 3/8″] gauge). Cf. figure 255.

Stahlguss = Cast steel. Anticorodal = Well known aluminium alloy.

If we have dwelt to some extent on flexible wheels, it is because we consider them — at least for certain purposes, subject to future experience — as having immense influence in the matter of *unsuspended weight*. It is, moreover, the tendency to reduce unsuspended weight which is one of the original and fundamental purposes of flexible transmission arrangements.

To complete the question of flexible wheels we show in figure 414 a recent application of the SAB wheel for tramways. For some time the rubber discs for SAB wheels have been vulcanised on plate which simplifies the machining of the metal discs and also the assembly of the wheels.

We may mention to conclude what concerns this Chapter II, some other recent applications in Europe, of *nose-suspended motors* for heavy traction. In America, and particularly with regard to the railways of the U.S.A., we have already mentioned the special case of heavy rails (257), which modifies conditions, since much heavier rails (in the region of 80 kg/m [176 lbs per metre] are obviously less susceptible to non-suspended weight. In particular, the heavy multi-unit Diesel-electric locomotives (on the main lines of the West between Lake Michigan and the Pacific) are fitted with nose-suspended motors. On the main electrified systems, however, such as the Pennsylvania and NYNH & H (single phase A.C.) and C.M. St-P. & Pacific (3 000 V D.C.), etc., the tendency to suspend the entire weight of the motors appears well defined.

On the line electrified experimentally in 1950-1951, with single phase high tension and at industrial frequency, from La Roche-sur-Foron to Annecy and Aix-les-Bains, the SNCF has put into service a certain number of motive units (locomotives and motor coaches) as prototypes or purely experimental.

Amongst these motor units (258) four have nose-suspended motors, viz :

1) B_0B_0 locomotive, No. 8051, with rectifiers, built in the Als-Thom Works (D.C. traction motors, 675 V, terminal voltage); max. speed in service : 105 km (65 miles)/h. Hourly rating : 2 840 HP; tare : 79 tons.
2) $B_0B_0B_0$ locomotive, No. 6053, with single phase/D.C. converter group, built by the « Matériel Electrique » Co. S-W (D.C. traction motors, 675 V terminal voltage); max. speed in service : 100 km (62 miles)/h (single phase of D.C.); hourly rating 3 820 HP; tare : 114 tons.
3) Twin motor coach Z 9053/54, with direct motors (single phase 50 cycles) built in Germany; max. speed in service : 105 km/h; hourly rating : 1 360 HP (1 920 respectively) (four motors, two per coach or four driving axles in a total of ten).
4) Motor coach Z 9055, ignition type, built by Westinghouse Electric; maximum speed in service : 80 km (50 miles)/h. Hourly rating : 760 HP (four driving axles in 8 or motor coach plus driving trailer).

With regard to other stock (259) on

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(257) See *Congress Bulletin*, May 1949, p. 383 and note (177) resp., and p. 221 of Vol. II. — American motive units (electric and Diesel-electric traction); see also *A.F.A.C.*, No. 170 (September/October 1951): « Le Chemin de Fer Américain en 1951 », D. CAIRE, particularly pp. 105-110.

(258) See *R.G.C.F.*, November 1951, pp. 701-704 and 708-713. — Owing to lack of space, illustrations of these are not included; the *R.G.C.F.* can be readily consulted. Furthermore, the motor suspensions have no new features. With regard to 2), see also *Bulletin de la Société française des Electriciens*, September 1951, 10 pages, 6 figures, ROSSIGNOL.

(259) In the *Congress Bulletin* of January 1950 (and Vol. II), SNCF locomotives 6051 (fig. 365) and 6052 (fig. 364) only were mentioned.

trial on the Aix-les-Bains-Annecy-La Roche-sur-Foron, these will be mentioned in dealing with the classes appropriate to their transmission types.

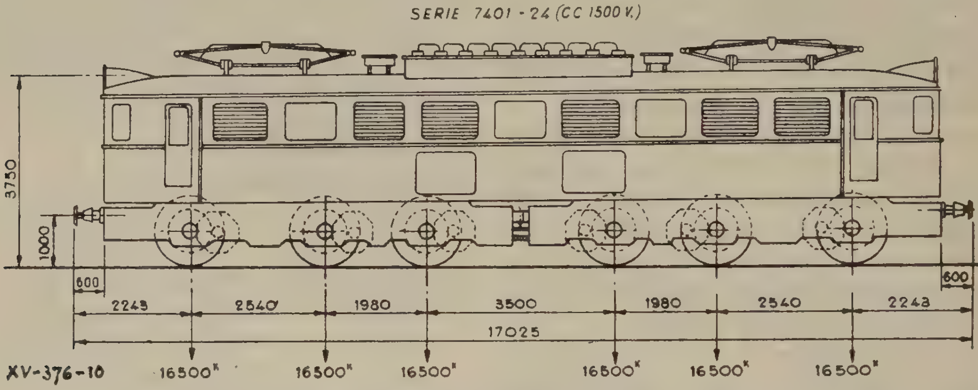
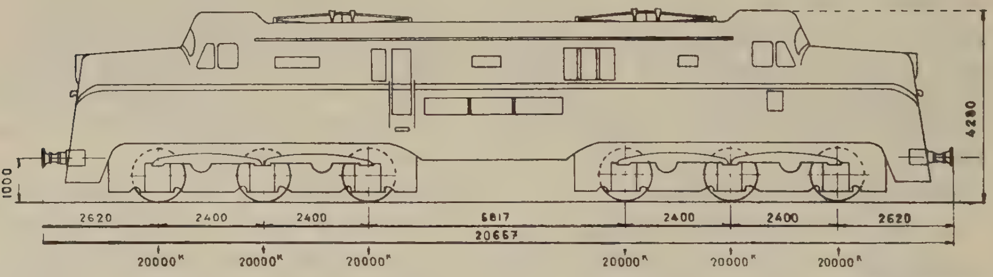


Fig. 415. — Dimensioned sketch showing arrangement of axles of 24 series 7401 locomotives of the Spanish Railways (RENFE), type C_0+C_0, 1 500 V, D.C.



Plans RENFE

Fig. 416. — Dimensioned general arrangement sketch of axle suspension for 60 C_0-C_0 locomotives, series 7701 of the Spanish RENFE, 3 000 V, D.C.

We may also mention briefly another recent prototype machine of the SNCF, the $B_0B_0B_0$ locomotive, No. 6002, 1 350 V, D.C., put into service in 1950 (260). This is a locomotive with a unique type of body, 17.79 m long, on 3 bogies each with two driving axles, the two outer bogies having on the middle bearer a pivot recoil arrangement (261).

The special nose-suspension of the motors of this locomotive is characterised by a specially designed rubber rim mounted on the knuckle (262). The gear wheels (bilateral drive, two gears per axle) have flexible rims with pivoted coil springs, similar to those of figure 405.

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(260) See *R.G.C.F.*, July 1950, pp. 301-317, 23 figures, « La locomotive électrique prototype type B-B-B-6002 SNCF », THOMACHOT & ROSSIGNOL, 23 figures and diagrams.

(261) See fig. 7, p. 306 of article in note (260).

(262) See fig. 14, p. 311 of article in note (260).

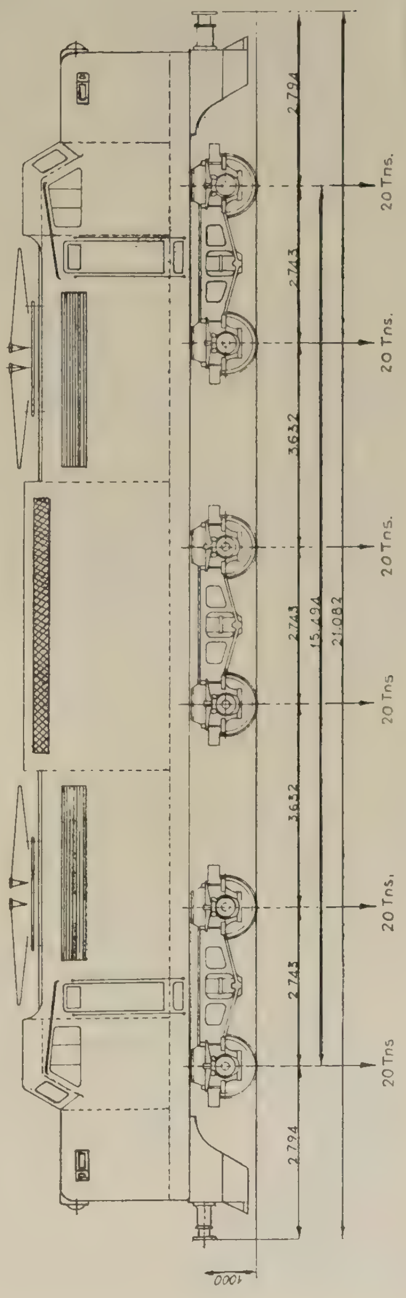
A flexible rim for the gear-wheels is in fact essential in the case of bilateral drive with nose-suspended motors.

The Spanish Railways RENFE (Red Nacional de los Ferrocarriles Españoles) have ordered during recent years (since the latter part of the second world war) a certain number of locomotives with nose-suspended motors, which it is interesting to mention here, in view of the large series. Apart from eighty C_0-C_0 engines with flexible transmission, series 7600 (Als-Thom), which will be mentioned in Chapter III, there are three series with nose-suspended motors, i.e. :

a) series 7400, comprising twenty-four C_0+C_0 locomotives, two bogies coupled, each with 3 axles. The six traction motors are inclined towards the centre of the locomotive. The gear wheel, with straight teeth, is keyed to the axle shaft. These twenty-four machines were put into service between 1944 and 1950; 1 500 V, D.C. (fig. 415);

b) series 7700, comprising sixty locomotives, the first two of which were put into service in the summer of 1950, type C_0-C_0, two independent bogies, with three driving axles each; 3 000 V, D.C. (fig. 416) (263);

c) series 7800, comprising twenty locomotives, also for 3 000 V, D.C., type B_0-B_0-B_0, three independent bogies, with two driving axles each (fig. 417).



Plan RENFE

Fig. 417. — Dimensioned sketch showing arrangements of three driving bogies of twenty $B_0B_0B_0$ locomotives, series 7801 of the Spanish RENFE. 3 000 V, D.C.

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(263) Compare the suspension of these locomotives with that of series 6001 electric locomotives, built about 1928, by GECo and the American Locomotive Company for the Rampa de Pajares (Leon to Venta de Baños, Gijon-Madrid) Spanish Northern Railways (NORTE), now Zona NORTE, RENFE.

See *La Traction électrique*, Pontoise-Paris, March 1931, pp. 76-77 (figs. 14 and 16).

The principal features of these three series are indicated in the table below :

Series (No.)	Nominal line voltage V	Adhesive weight = total weight t	Gear ratio	Wheel diameter mm	Maximum speed km/h	Hourly rating (tread) HP	Tractive effort (at tread) kg	At a speed of km/h (*)
7400 (24)	1 500	99	4.94	1 300	100	3 000	17 100	48
7700 (60)	3 000	120	3.94	1 220	110	3 560	17 400	55
7800 (20)	3 000	120	3.15	1 118	130	3 000	16 700	43

(*) At 100 % field.

The transmission (gears) is mainly single sided. Series 7400 and 7800 have rigid pinions and gears, whilst series 7700 has flexible gears as in figure 418, which it is interesting to compare with figure 388, page 784, *Congress Bulletin* of January 1950, p. 45 (last figure of Vol. II).

The companies which built or are building these locomotives are :

Series	Mechanical parts.	Electrical part.
7400	M.A.C.O.S.A., Valencia	Sécheron, Geneva
7700	Vulcan Foundry and English Electric Co	English Electric Co
7800	Sdad. Española de Construcción Naval	Westinghouse Electric, U.S.A.

We may also note that on series 7400 the combination of motors is series or series parallel, whilst for 7700 and 7800 there are series, series parallel and parallel. All of them have in addition full field and two shunt tappings.

Finally, the new 1 500 V electric motor coaches for RENFE, at present under construction, to the number of 15, also have nose-suspended motors and are almost identical (apart from some mechanical modifications) to the former 300 series, put into service about 1930 on the Barcelona to Manresa and St. Juan de las Abadesas and San Sebastian to Irun lines (264)

The Netherlands State Railways, NS, have also had built a series of twenty-five C_0-C_0 locomotives with nose-suspended motors. These are the 1200 series, Nos. 1201-1225, a markedly American type for a maximum speed in service of

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(264) See *La Traction électrique*, Pontoise-Paris, Nos. 1-2, Vol. I (December 1929-January 1930), pp. 14-15, figs. 4 and 5.

Also see *E.T.T.* (formerly *L'Allègement dans les transports*), Nos. 7-8, 1933, pp. 86-87, fig. 5 : « Les procédés d'égalisation des charges dans les bogies », J. WORMS and M. BONNET.

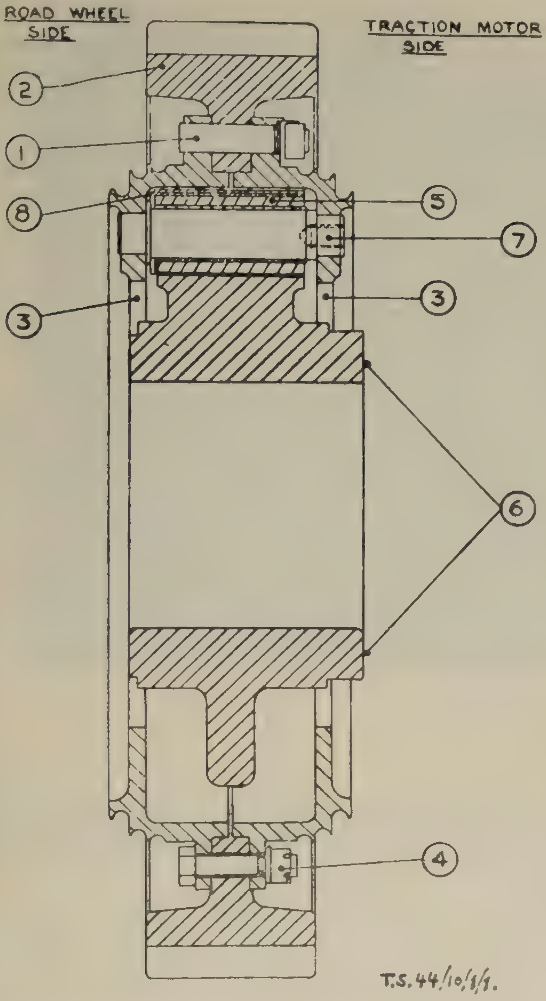

Fig. 418. — Flexible gear wheel of electric locomotives Nos. 7701 to 7760 (1950) of the Spanish RENFE :

1 — pegs, diametrically opposite bolts 4.'
2 — geared rim.
3 — rims located round the gear wheel centre (6) fixed by the pegs (1) and the bolts (4).
4 — bolts, diametrically opposite pegs (1).
5 — flexible sleeves of silent-block type.
6 — gear wheel centre.
7 — key.
— sealing rings.

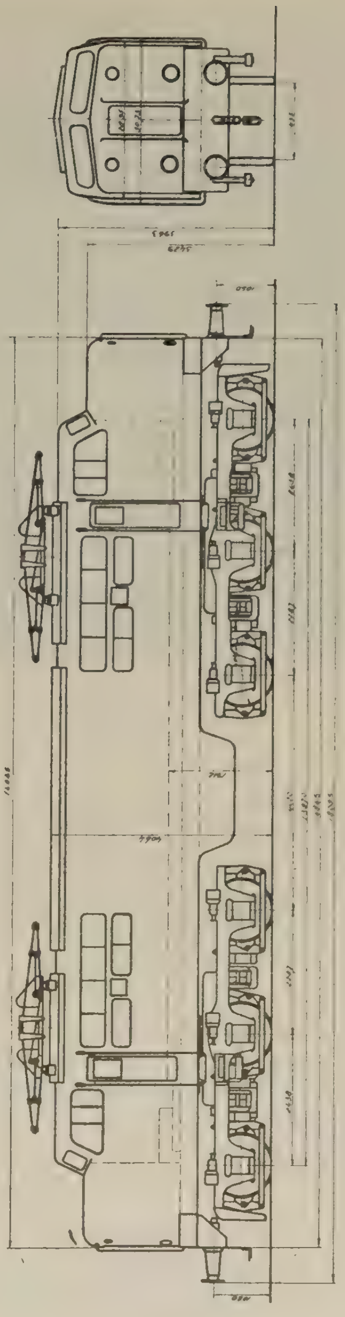

Fig. 419. — Dimensioned sketch of C_0-C_0 locomotives, series 1201, Netherlands Railways (N.S.)

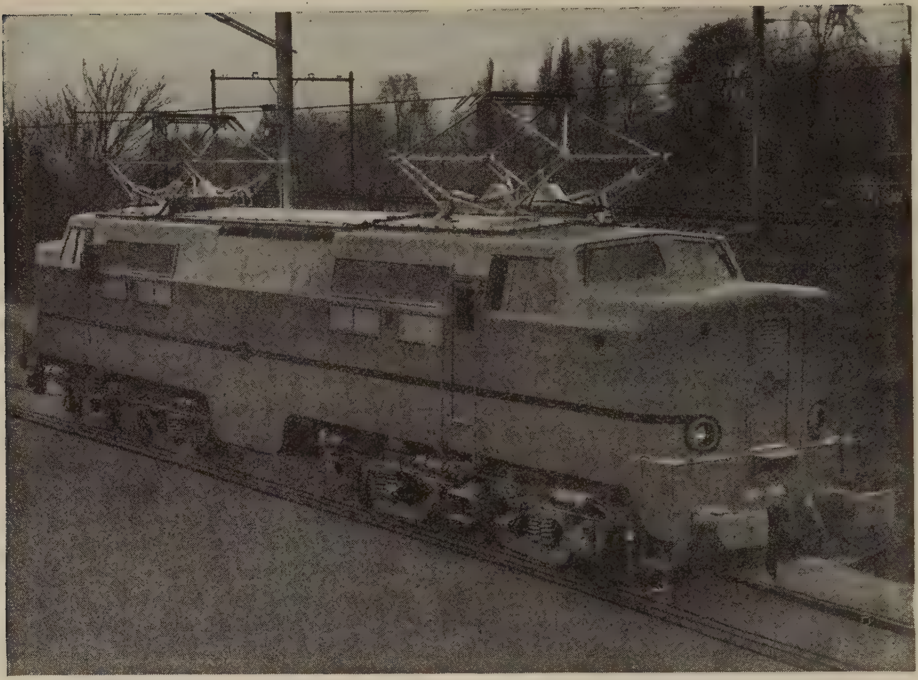

Fig. 420. — A locomotive, figure 419, on the line.

140 km (87 miles)/h, hourly rating of nearly : 4 000 HP (at 1 500 V). The manufacturers were, for the mechanical parts, Baldwin and Werkspoor and for the electrical equipment, the Westinghouse and Heemaf Works (265).

These machines, shown in figures 419 and 420, have the peculiarity that the axles, with roller bearing boxes, are directly driven through non-resilient spur gears by the traction motors (which are nose-suspended); there is consequently practically *no flexibility* in the wheel arrangement of these locomotives. Figure 421 shows the arrangement of the motors in the bogies.

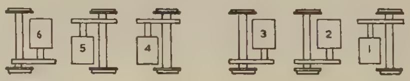

Fig. 421. — Arrangement of traction motors (rigid) of locomotives in figures 419 and 420.

Apart from the dimensions shown in figure 419, we may note that the wheel diameter (new) is 1.10 m (3′ 7 5/16″) and the gear ration 20 : 71. The tare (= adhesive weight) of these machines is 105 t which corresponds to an axle load of 17.5 tons average. The continuous rating, at 72 km (44 miles)/h for wheels worn to 1 060 mm (3′ 5 6/8″), at 1 350 V, is 3 000 HP, or a maximum hourly rating at 1 500 V of about 3 500 HP. Finally, the speeds in service are : *normal*, 125 km (77 miles)/h; maximum : 140 km (87 miles)/h with 12 notches of economical running.

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(265) See : *Congress Bulletin T.E.*, June 1952, pp. 97-107, 9 figures;
A.F.A.C., No. 166, 1951, pp. 9 and 10;
De Ingenieur, No. 2, January 12, 1951, 8 figures, 4 tables and graphs.

Any comparison made by the NS (after a period in service) between this series 1 200 and the 10 C_0-C_0, Als-Thom type (series 1300) should be of considerable interest.

The motor coach sets of the NS (electric and Diesel-electric) have all motors nose-suspended. Many descriptions of these sets have been published (266).

CHAPTER III.

DRIVING MECHANISMS WITH PUSH-RODS AND ARTICULATED JOINTS.

A. — Buchli mechanism (267).

Among new applications, there is only the type *b* (*Congress Bulletin* of October 1947, figs. 13-15 = p. 21 of Vol. II) as in fig. 15, bilateral drive without hollow shaft, mechanism on both sides, *external* to the axle.

When the Paris-Lyons line was electrified (direct current with average line tension of 1 350 V), the SNCF had to decide on suitable locomotives for fast heavy trains of 700 tons (16 coaches and 2 brake vans) at speeds of 140 km (87 miles)/h and later even of 160 km (100 miles)/h. This consequently necessitated an adhesive weight of 92 tons (distributed on 4 driving axles), giving an hourly rating of 5 000 HP which corresponds to an hourly tractive effort of about 22 tons at a speed of approx. 62 km (38 miles)/h. The maximum speed in service should in fact be 160 km/h, which requires a tractive effort at rail or minimum field (at 1 350 V) of 6.6 tons (almost 4 000 HP).

These figures could already be achieved (at least very closely) by the five 2-D_0-2 locomotives Nos. 5546-5550, put into service in 1942 (268). The SNCF subsequently ordered from the Brown-Boveri Company of Switzerland two prototype machines Nos. 9101 and 9102 (269) of the same general arrangement as the very first (Nos. 501 and 502) put into service in 1925/26 by the P.-O. Railways (270). These new machines have however been redesigned to take advantage of the experience acquired during twenty-five years service; these factors are :

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(266) The most recent Diesel electric sets are described in *Spoor- en Tramwegen*, The Hague/Utrecht, *Nieuw dieselelectrisch reizigersmaterieel voor de N.S.*, 20 March, 3 April, and 17 April 1952, 9 pages, 16 figures, J.P. Koster (see *Congress Bulletin T.E.*, December 1952, pp. 275-291.

(267) As previously, we take first the oldest type (used in 1920) and which has recently, at least in France, had a fairly wide use; see Vol. I, pp. 22-35, figs. 38-70, and Vol. II, pp. 18-24 and 224-225, figs. 13-21 (*Congress Bulletin*, October 1947 and May 1949).

(268) See *Congress Bulletin*, May 1949, p. 385 (p. 224, Vol. II).

(269) See *R.G.C.F.*, Paris, June 1950, pp. 253-267, 26 figures : « The 2-D-2 locomotives of the Paris-Lyons line », J. Woimant.

See also *A.F.A.C.*, No. 162, May/June 1950, pp. 49-58, 19 figures : « The evolution of the 2-D-2 locomotive from the Fives-Lilles Electro-Mechanical Co, from E 501 to 2-D-2 9101 », A. Gache and D. Caire.

1) to increase by a third the power of the machine (in continuous rating, 4 400 HP instead of 3 320);
2) to increase the maximum speed in service to 160 km (100 miles)/h (instead of 130 to 150 km [80 to 93 miles]/h as for the earlier machines);
3) to increase — by taking advantage of the new permissible axle load of 23 tons on main lines — the adhesive weight, also by nearly a third (from 72 tons, then 80, to 92 tons) (271).
4) to utilize to better advantage the adhesion (starting and shunting) by the provision of a much greater number of notches.

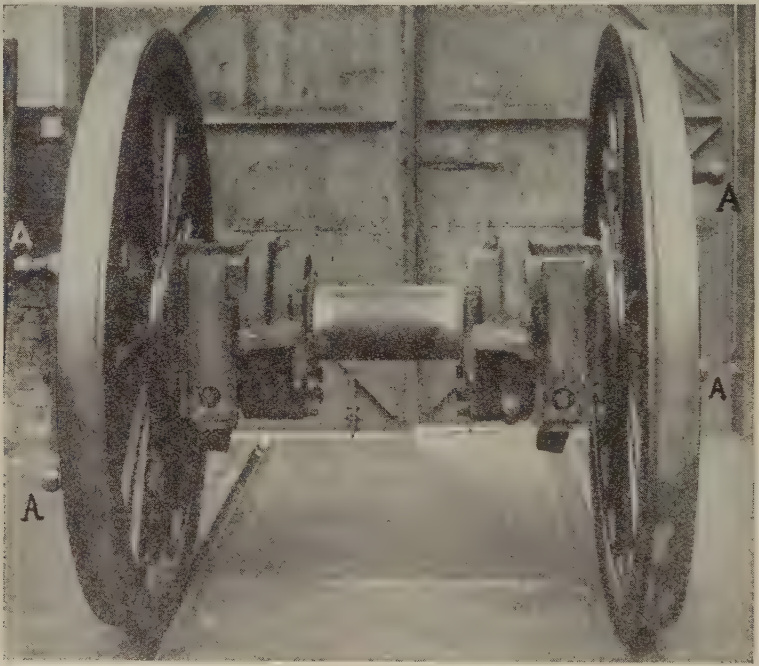

Fig. 422. — Driving axle of SNCF 2-D$_0$-2 locomotives, series 9101 (similar to fig. 52, p. 28, Vol. I) showing the pivots A for the push rods. The pivots are diametrically opposed at each side. Also to be seen are the solid discs of the wheel centre and the inside Athermos boxes (type F 5^3 for 240 × 327 mm [7′ 10 1/2″ × 10′ 8 3/4″] journals) (some hundreds of this type are in service), the operation of which is shown in figure 279.

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(270) In addition to the two machines to this 2-D$_0$-2 prototype, the SNCF at the same time ordered from another contractor two other locomotives B$_0$-B$_0$ prototypes and with these we shall deal later. See also preface to article in note (264).

(271) For running over less robust lines, the axle load can be reduced from 23 to 21.5 tons by dispensing with 2 tons of ballast and redistributing the load on the driving and carrying axles by adjusting the suspension.

The principal characteristics of the new prototype are as follows (272) :

— total length over buffers	m	18.080 (59′ 3 25/32″)
— total wheelbase	m	14.690 (48′ 2 1/8″)
— rigid wheelbase of outer driving axles	m	6.150 (20′ 2 5/32″)
— diameter of driving wheels	m	1.750 (5′ 8 7/8″)
— diameter of carrying wheels	m	1.000 (3′ 3 3/8″)
— weight of mechanical part	t	89.6
— weight of electrical equipment	t	56.6
— weight of accessories	t	1.8
— ballast	t	2.0
— total weight in working order	t	150
— adhesive weight	t	92
— load per driving axle	t	23
— load per outer carrying axle	t	13.3
— load per inner carrying axle	t	15.7
— minimum curvature (at low speed)	m	80
— normal line tension at pantograph	V	1 350
— maximum line tension	V	1 800
— continuous rating at 1 350 V	HP	4 400
at a speed of	km/h	65.3
corresponding to a tractive effort of	kg	18 240
— hourly rating at 1 350 V	HP	4 880
at a speed of	km/h	62.5
corresponding to a tractive effort of	kg	21 100
— gear ratio		1 : 2.31

These locomotives were built by the Compagnie Electro-Mécanique (electrical equipment) and the Compagnie de Fives-Lille (mechanical parts).

To this model (locomotives 9101 and 9102) thirty-three similar locomotives, Nos. 9103 to 9135 have been built and put into service successively since 1950, which brings the total of 2-D-2 locomotives of this type with Buchli mechanism on the SNCF to 73 + 35 = 108 locomotives (273).

Figure 422 shows a driving axle of these new locomotives; full wheel disc, with the two outer pivots A to each wheel and the forced lubrication boxes, necessarily located inside (274). The carrying axles are similar to those of figure 280, page 385, *Congress Bulletin* of May 1949 (275).

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(272) Compare this with the table on page 385 (*Congress Bulletin* of May 1949 — p. 224 of Vol. II).

(273) There are of course many other 2-D-2 machines on the SNCF fitted with other mechanisms. See for examples figs. 22 and 30, 104, 107, 108-110, 281, etc. (*Congress Bulletin*, 1947-1948-1949).

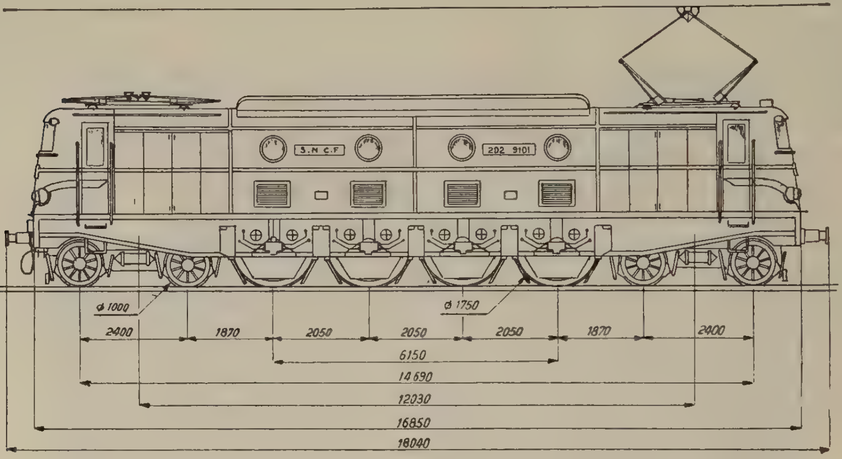
(274) See figs. 50-53, pp. 27-28 of Vol. I and figs. 13-15, pp. 18-20 of Vol. II (*Congress Bulletin* of October 1947).

(275) p. 223 of Vol. II.

Figures 423 and 424 respectively show a dimensioned sketch of the 9101 series and one of these locomotives hauling an express train.

As far as we know, there have been no other applications of the *Buchli* mechanism during the past four or five years.

We may also mention — in connection with the present putting into service (1951-52) of a large number of engines of the 2-D_0-2 type — the question of the number of *carrying* axles of a locomotive, which is important in relation to the individual driving of axles, and therefore of general interest in the development of systems, particularly electric traction. We know that the load on the carrying axles of locomotives is considered to be dead weight since it does not contribute to the adhesion. In most types of locomotives, particularly those of high power and speed, of the last decades, carrying axles were necessary, either to assist in carrying the load of electrical equipment (which was too great for the carrying capacity of the driving axles) or for guiding the locomotives at high speeds, or for both purposes jointly.



From A.F.A.C.

Fig. 423. — Dimensioned sketch of the thirty-five SNCF 2-D_0-2 locomotives, series 9101, Brown-Boveri-Buchli mechanism (bilateral, outside). (Cf. fig. 50, p. 27, Vol. I.)

Meanwhile, however, in addition to the tendency to increase permissible axle loads, there has also been progress in the fabrication of motors and equipment which has permitted a large decrease (up to and even exceeding one half) in the weight to power ratio. For these reasons, railways have tended to adopt more and more the total adhesion locomotive, particularly the B_0B_0, C_0C_0 and B_0-B_0-B_0 types (276).

The type of machine which at present

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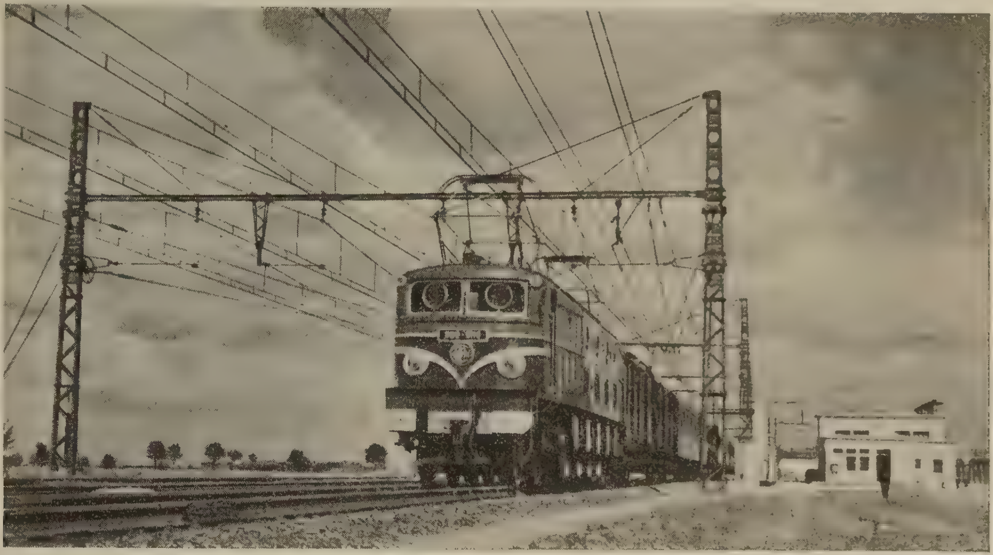
(276) See *R.G.C.F.*, February 1947, « Le comportement en marche des locomotives électriques à grande vitesse », Ad.-M. HUG, 6 pages, 3 figures.

seems to give the best axles guiding, either on curves or straight track (with a carrying axle at each end, however, contained in a mixed bogie with the next axle driving) is still the express electric locomotive type 1A-AA-A1, series 3001 of the Indonisian State Railways D.K.A. « Djawatan Kereta Api » (formerly the S.S. — Staatsspoorwegen in Nederlands Indië — up to 1945, when sovereignity passed to the Indonesian Republic). The six locomotives of this type, the first two of which were put into service in 1924/25 on the Djakarta (Batavia) district electrified lines, were described in 1931 (277). They are fitted with the Brown-Boveri-Buchli drive.

As the articles published date from 1928-1931-1933, or twenty years and more ago, it may be worth reproducing — in figure 425 — the frame (with dimensions) of these machines, showing the arrangements of the axles and pivots. These combined bogies, known as the « Java » type (because they were first used in Java) have the pivot located outside the wheelbase, towards the centre of the locomotive. Similar bogies have been used :

— since 1927, on twenty-two locomotives,



C.E.M. n° 25572 Photograph.

Fig. 424. — 2-D_0-2 locomotive No. 9110, hauling an express between Paris and Lyons, passing a substation (cf. fig. 423 and also figs. 13 and 15).

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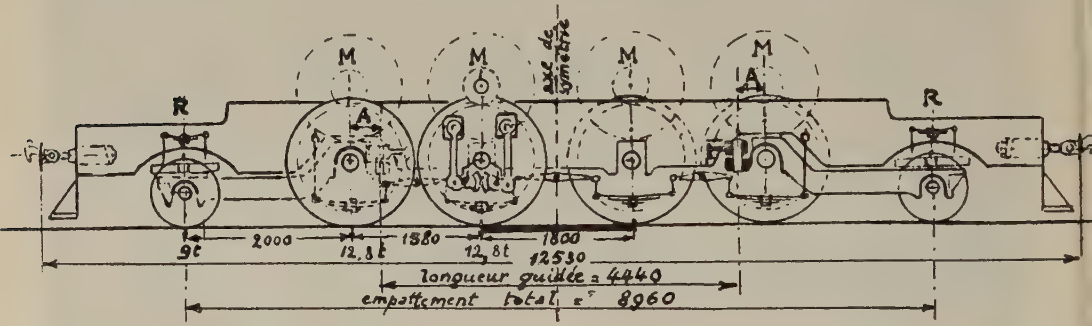
(277) See Vol. I, p. 26 (figs. 46-49) and *La Traction électrique*, Paris, No. 7, 1931, pp. 176-177 (figs. 45-48).

Also : *Bulletin de la Société Belge des Ingénieurs et des Industriels*, Brussels, February-March 1929, pp. 144-145; *R.G.C.F.*, May 1928, p. 378; *E.B.*, November 1928, p. 339-343.

type 1A-AA-2 (No. 4000, prototype, and Nos. 4004-4024, class EA/1) of the Great Indian Peninsula Railway, G.I.P., now part of the Indian Railways, I.R. (^{278});

- since 1933-1934, on a certain number of SBB — CFF — FFS (Swiss Federal) locomotives, but with the pivot inside the bogie wheelbase, near the driving axle : since 1931 on the superlocomotive Ae$^{8}/_{14}$, No. 11851, and since 1934 on the Ae$^{4}/_{7}$ locomotives of the series from No. 10973 (^{279});
- finally, about 1930, various locomotives in Japan (J.G.R., series 7001, 2 200 HP, 100 km [62 miles]/h, 1 500 V, D.C.) and in Brazil (Paulista C.P., prototype No. 231, 3 100 HP, 105 km [65 miles]/h, 3 000 V, D.C.) (^{280}).

To conclude this section on the Brown-Boveri-Buchli mechanism, we may refer briefly to the eighteen locomotives, 2-C_0+C_0-2 type, series 7200 and 7500 of the Spanish Railways RENFE (resp. RN) — Red Nacional de los Ferrocarriles Españoles (^{281}). These as we know have Buchli bilateral drive, but with mechanism



From R.G.C.

Fig. 425. — Frame and wheels of electric locomotives for express trains, type 1A-AA-A1, series 3001 of the Indonisian DKA Railways. Hourly rating : 1 600 HP; maximum speed in service : 90 km (56 miles)/h, 1.067 m (3′ 5 1/2″) gauge. The distance A from the bogie pivot to the near driving axle centre line is only 26 cm (10 1/4″), which allows free play for the axle in transmission of the tractive effort by means of the individual drive mechanism, the four motors being mounted in the main frame; moreover, the distance is in this particular case less than a quarter of the rail gauge, which gives better stability for axle control than the bogie wheelbase would ensure.

M = Motors. — R = Recoil arrangement between bogie and frame.

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(278) See pp. 6-10 (figs. 4-10) of *La Traction électrique*, Paris : « Les locomotives électriques à grande vitesse du GIPRy ».

See also Vol. I, pp. 60-61 (figs. 126-130). — These twenty-two locomotives as well as No. 11851 of the CFF do not have Brown-Boveri-Buchli transmission; they have the SLM-Winterthur « Universal » type, both having however been constructed by the engineer J. BUCHLI.

(279) See *Congress Bulletin*, October 1947, p. 887 (p. 19 of Vol. II). — For the Japanese and Brazilian locomotives : Vol. I, figs. 58-60 and 62-63.

(280) See Vol. I, pp. 30-31 and figs. 58-60 and 62-63.

(281) See figs. 54-57, pp. 28-39 of Vol. I (regarding the first series 7.201) and fig. 21, *Congress Bulletin* of October 1947, p. 894 (p. 26 Vol. II for second series, 7501). — *Errata* : in the legend of the latter fig. 21, p. 26 (II) read series 7.501 not 3.501 (see Vol. II, p. 312).

inside (caused largely by the wide gauge of 1 674 mm [5′ 5 29/32″]) in relation to the rolling stock gauge and the width of the bodies), which entails a hollow shaft along the full length of the axle between the wheels.

The electric traction division of the RENFE state in regard to this question that failures owing to flexible pinion fractures in the Brown-Boveri-Buchli transmission are frequent. If we are correctly informed, there are also difficulties experienced with the spherical pivot heads (A, in figure 422, but inside, see fig. 56, p. 29, Vol. I) because of the infiltration of dust and grit (and rubbings of iron from the brake blocks). The RENFE is considering the possibility of modifying the mechanism [possibly using rubber pads, somewhat similar to the arrangement in figure 336 (p. 798 of the *Congress Bulletin* of November 1949) (282)], to eliminate these defects, which are making operation difficult. The question is under investigation. It must be borne in mind that these locomotives have been in continuous heavy service for more than a quarter of a century and have run millions of km. Moreover, bilateral mechanism inside the axle is very inaccessible and maintenance is therefore difficult.

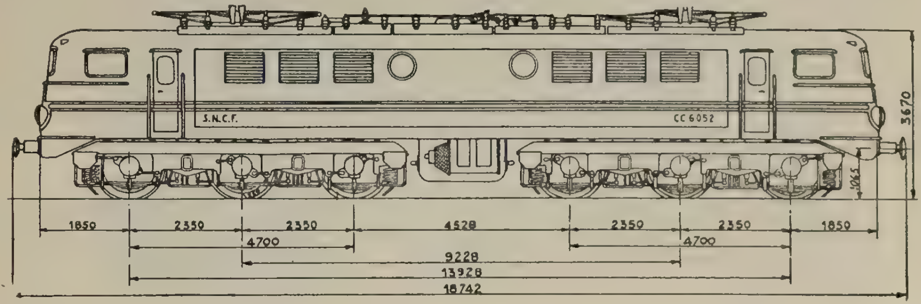

Fig. 426. — Dimensioned sketch of 50 cycle single phase locomotive No. 6052, type C_0-C_0, SNCF. Finalised version of figure 364.

Finally, we have already mentioned on p. 892 of the *Congress Bulletin* of October 1947 (283) (according to the *S.E.V. Bulletin*, Zurich, Swiss Electrotechnical Association, 1943) that the mechanism was obsolete and that it was stated there would be no new applications. There are, however, over the world not far from 500 locomotives with Buchli mechanism, the earliest having been in service since 1920.

The mechanisms under B *(Oerlikon I and II)* and C *(Skoda)* having, so far as we are aware, no further applications (see *Congress Bulletin* of October 1947, p. 893) (284), we will pass to developments of the « Alsacienne » and « Als-Thom » mechanisms (see *Congress Bulletin* of October 1947, from p. 893) (285).

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(282) p. 265 of Vol. II.
(283) p. 24 of Vol. II.
(284) p. 25 of Vol. II.
(285) Vol. I, p. 74, figs 154-156; Vol. II, pp. 25-34 and 225-232.

D. — Als-Thom mechanism.

This flexible transmission was originally that of the Société Alsacienne de Constructions Mécaniques (Belfort), shown in figure 154 of Vol. I (285). Its present appearance is practically the same as that in the illustrations in the *Congress Bulletin* of October 1947, from p. 896, figs. 23, 25 and 29, then figs. 285-286 and 290-291. *Congress Bulletin* for May 1949 (286).

Amongst the applications so far mentioned — they were only under construction or on order in 1949 — there are fifty B_0-B_0, series 1101, locomotives of the Netherlands (NS) Railways (fig. 292) and the 50 cycle single phase C_0-C_0 locomotive, No. 6052, of the SNCF, South East Region (fig. 364) (287).

We shall later describe the 50 B_0-B_0 locomotives 1101-1150 of the NS and the 10 C_0-C_0, series 1301, whilst figure 426 includes a dimensioned sketch of the SNCF locomotive, No. 6052 (which differs somewhat from fig. 364); figure 427 shows a group of two twin motors of one axle with the two transmission pivots and figure 428 a bogie with six motors (3 × 2 twin). The general appearance of the machine is similar to that of figure 282.

Below is a table of the various series since put into service, or ordered, with Als-Thom mechanism, in order of their appearance :

	SYSTEM	Type	Series No.	Year of introduction into service	Remarks
1	NS — Netherlands	B_0'-B_0'	1101-1150	1950	50 locomotives in the series
2	SNCF — France (S. E. Region)	C_0'-C_0'	6052	1951	50 cycle single phase prototype (fig. 426)
3	NS — Netherlands	C_0'-C_0'	1301-1310	1952	10 locomotives in the series
4	SNCF — France (S. E. Region)	C_0'-C_0'	7101-7153	1952	53 machines in the series
5	RENFE — SPAIN	C_0'-C_0'	7601-7680	1952	80 machines in the series (fig. 429)

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(286) pp. 229, 230 and 232, and 232-233 of Vol. II.

(287) See publication mentioned in note (258), pp. 696-700.
Also *A.F.A.C.*, No. 170, September/October 1951, pp. 113-116.

The total to date with the machines mentioned previously since 1935, is 202 locomotives and 2 double motor coach sets (Z 4, 701 and 702, see figs. 284-290).

Figure 429 is a dimensioned sketch of the eighty C_0-C_0 locomotives, Nos. 7601-7680 (item 5 of above table). These are at present under construction, the first

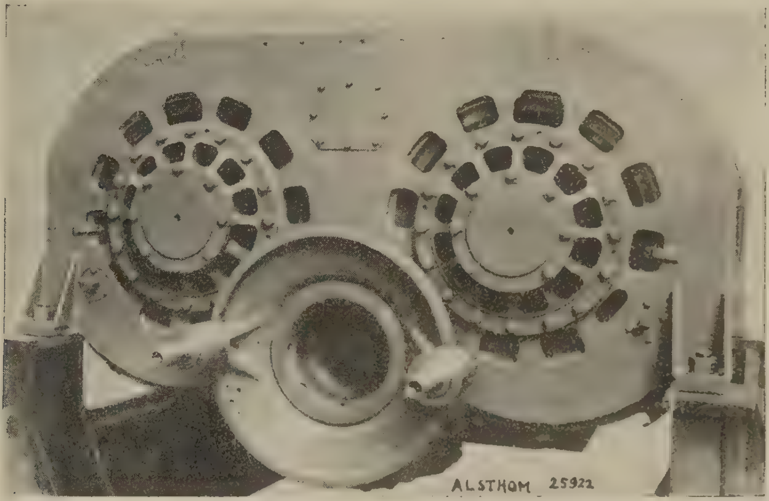


Fig. 427. — Twin single phase (50 cycle) motor with — mounted in its cage — the hollow shaft fitted with two pivots for the Als-Thom drive mechanism. SNCF locomotive, type C_0-C_0, No. 6052.

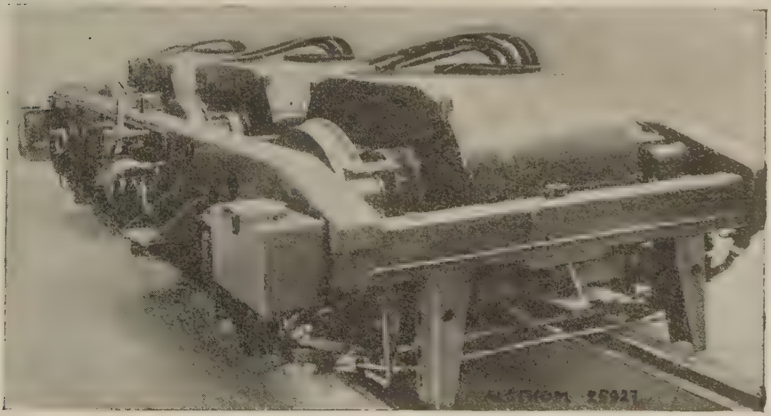


Fig. 428. — Motor bogie complete for locomotive shown in figure 426 (cf. fig. 427).

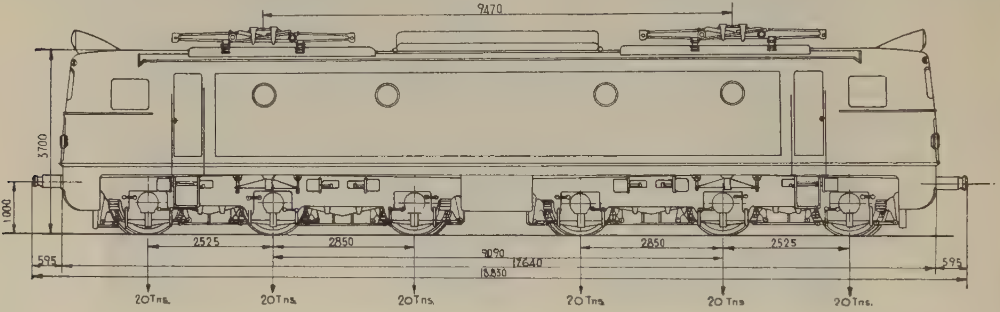
20 in France (Als-Thom, mechanical and electrical parts) and the next 60 in Spain, the contractors being as follows :

Mechanical parts :

Material y Construcciones S.A., Valencia and Barcelona;

Compañia Auxiliar de Ferrocarriles, Beasain;

Babcock and Wilcox, Bilbao-Galindo;



Plan RENFE XV. 357.3

Fig. 429. — Dimensioned sketch of Spanish RENFE locomotives, series 7601, C_0-C_0 type, for 3 000 V, D.C. similar to prototype 7001/2 and SNCF, series 7101.

Maquinista Terrestre y Maritima, Barcelona;

Compañia Euskalduna de Construcción y Reparación de Buques, Bilbao;

Electrical equipment :

General Eléctrica Española, Galindo (Bilbao);

Constructura Nacional de Maquinaria Eléctrica S.A., Reinosa (Santander);

Placencia de las Armas S.A., Placencia (Guipuzcoa), and Experiencias Industriales, Aranjuez (Madrid);

Sociedad Ibérica de Construcciones Eléctricas, Madrid;

Assembly :

General Eléctrica Española, Galindo.

These locomotives will be practically identical with the prototype No. 7001 and series 7101 of the SNCF apart from the *track gauge*, which on main lines in the Iberian peninsula is 1 674 mm. We will therefore give only those details given for the 7400, 7700 and 7800 series, dealt with briefly in Chapter II.

The tare of these locomotives (= adhesive weight) is 120 tons; gear ratio : 3.15, diameter of wheels : 1 350 mm (4′ 5 1/8″), maximum speed : 125 km (77 miles)/h; hourly rating at rail : 3 150 HP and the tractive effort at rail varies, according to the percentage of field and speed, from 11 000 to 17 200 kg (24 250 to 37 920 lbs) at hourly rating.

(To be continued.)

[625 .251]

Confines of braking,(*)

by H. R. Broadbent, B. Eng.

PART 1. — Factors reducing braking rates below the limit imposed by adhesion between wheel and rail.

With torque braking, as distinct from track braking, the maximum force which can be produced in a direction parallel to the track is limited by the coefficient of friction between wheel and rail, *i.e.*, the adhesion. Many other factors reduce the braking of a train below the maximum, but if all these depreciating effects could be got rid of there would remain the adhesion between wheel and rail as the ultimate limit.

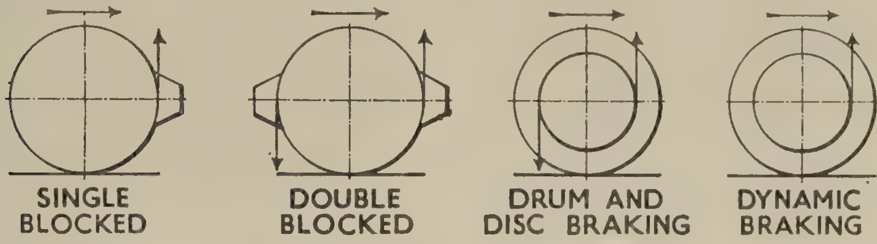


Fig. 1. — Methods of torque braking.

With torque braking, whether the torque is applied, figure 1, by the single force of a brake block at the wheel tread acting about the axle centre, or two forces with two brake blocks, or through a gearwheel, disc or drum, the effect is to produce, figure 2, a couple resulting in a force P_D parallel to the track, decelerating at a rate f the masses m of the car through the centre of mass of the wheel and axle assembly and through horn guides. An equal force P_T acts on the rails through the contact between wheel and rail with consequent reaction R_T on the wheel.

In order that there shall be equilibrium at the point of contact between wheel and rail, *i.e.*, that there shall be no relative movement, no wheel pick-up, no skid, the force P_T must be no greater than R_T, that is no greater than $W\psi$ where W is the weight on rail and ψ is the coefficient of friction between wheel and rail, the adhesion (as shown in fig. 3).

It is to be noted that the full torque on the wheel must allow for the deceleration of the rotating masses, but, with every axle braked, the additional torques, since

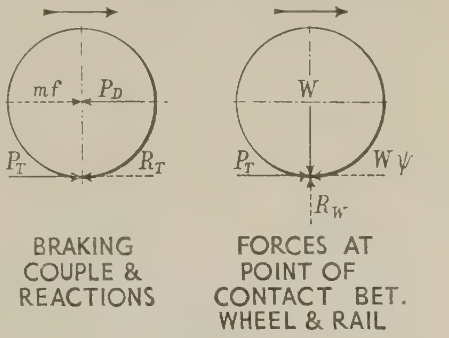

mf
P_D
P_T
R_T
BRAKING COUPLE & REACTIONS
W
P_T
$W\psi$
R_W
FORCES AT POINT OF CONTACT BET. WHEEL & RAIL

Fig. 2 (left) and 3 (right). — Forces involved in braking and requirements for adhesion.

(*) This article appeared in serial form in *The Railway Gazette* (see issues of Oct. 31, Nov. 7, Nov. 14, Nov. 21 and Dec. 5, 1952).

they decelerate the rotating masses without recourse to reaction at the wheel tread, may be ignored in the study of adhesion, except where the axle is overbraked.

It will be seen that since $P_D = P_T$

$$mf = W\psi$$

or, where each axle is arranged to brake the masses of which the weight on rail is the measure,

$$f = g\psi$$

i.e., the maximum rate of braking is controlled by ψ, the adhesion between wheel and rail.

It is common practice in discussing braking to refer to the « braking efficiency » of a train The figure, as a percentage, is derived from the equation — Braking efficiency % =

$$\frac{3{\cdot}34\ (\text{m.p.h.})^2}{\text{Braking distance in feet}}$$

The equation is derived from the kinetic energy formula :

$$Ps = \tfrac{1}{2} m V^2$$

$$\text{or } \frac{P}{W} = \frac{V^2}{2gs}$$

from which it will be seen that braking efficiency is the average braking force P parallel to the track during a braking application from a speed V to standstill, in distance *s*, expressed as a percentage of the train weight W. It will here be noted that since :

$$\frac{P_D}{W} = \psi$$

the braking efficiency can be related to adhesion, but only when the braking is straight line from moment of application to standstill. This is never so in practice, and the braking efficiency can therefore be considered only as an average ratio, and not the adhesion value which is the limit in maximum braking. It does, however, give an indication of the degree to which even emergency braking falls short of that which would be possible if continuous full use were made during a stop of the rolling coefficient of friction between wheel and rail.

Considerations of the other factors which limit braking must always be coloured by the fact that the highest rate which can be achieved is controlled by the coefficient of friction between wheel and rail. The number of factors which, for one reason or another, force the use of a braking lower than the highest rate is considerable. Each factor is treated separately hereafter, in the order shown in the following table, an indication being given of the manner in which it affects braking :

Limitation due to various factors :

(1) *Time.*
  Passenger comfort.
  Equipment response.

(2) *Apparatus.*
  Brake cylinder air pressure.
  Dynamic braking apparatus.
  Brake rigging efficiency.

(3) *Brake block.*

(4) *Block and wheel.*
  Variation of coefficient of friction (μ) of a brake block with speed.
  Block μ and applied force.
  Block μ and tyre temperature.
  Block μ and weather.
  Effect of wheel diameter with drum, disc and dynamic braking.

(5) *Wheel and rail.*
  Variation in passenger loading.
  Weight transfer.
  Deposits on rail.
  Effect of speed.
  Mutual wheel/rail contact.
  Gradients.

(6) *Train.*
  Frictional resistance.
  Windage.

(7) *Economic factors.*
  Range of air pressure.
  Effect on wheel and tyre of high rates of work stresses.
  Rate of wear of blocks and tyres.
  Standardisation of parts.
  Space.

Time factor.

It has already been mentioned that no braking starts at the maximum rate instantaneously, nor does it usually carry on to a stop at the maximum rate. The reasons are regard for passenger comfort and the speed of equipment response.

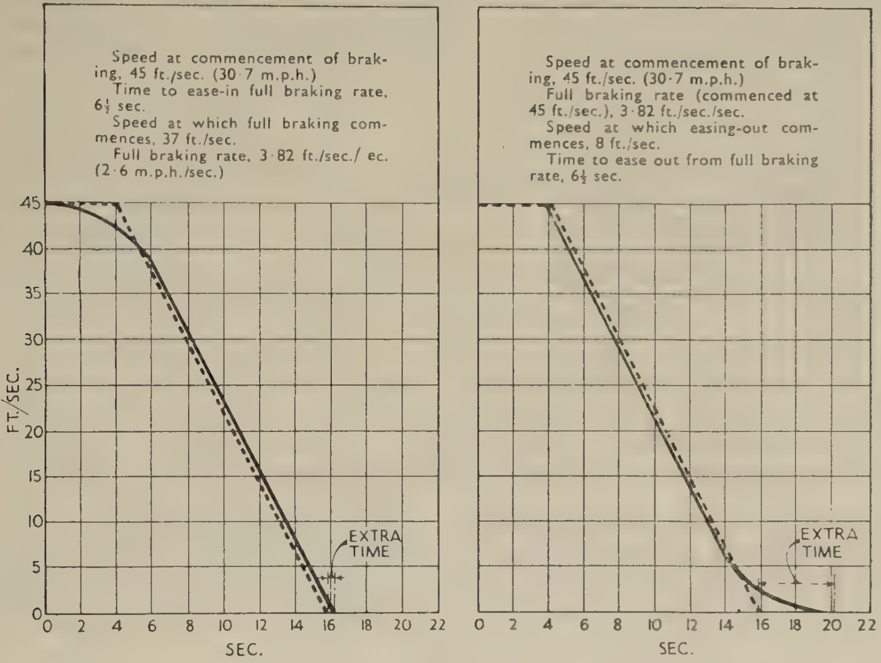


Fig. 4. — Effect of easing-in (left) and easing-out (right) on run-time.

Passenger comfort. — In service braking the driver tends to ease his brake into the maximum rate and out of it again towards the end of a stop. The perfect brake so far as the passenger is concerned is one of which he is unaware. The actual rate of braking, with a steel tyre on steel rails, can never reach values which could worry passengers. What does affect them is the rate at which the braking changes. This is the reason for easing-in and easing-out in service braking.

Figure 4 illustrates the point that easing-in with a service brake affects the run time hardly at all, whereas easing-out at exactly the same rate produces a major change, increasing run time by 4.16 sec. If schedule times are tight, one of the points on which an operator must concentrate is the end of the stops. Time can be lost unduly during this part of a run.

Equipment response. — To a driver the perfect brake is one which reacts instantly to his demands. Figure 5 shows a record of brake applications made with an eight-car stationary train to indicate the intervals between movement of the brake handle and the rise of air pressure in the brake cylinders. With a pneumatic brake, in which the entry of air to the cylinders is controlled by triple valves, there is an appreciable time lag even on the leading car of the train. It will be observed that the time to first movement of air in brake cylinder from first movement of brake handle increases from 2 1/4 at the front to 3 1/3 sec. at the rear of the train.

To the degree that a driver is unable to make the train follow his actions in braking there will be a deficiency for which the driver will automatically allow in his braking. The amount is indefinable, but if designers can cut down the lag in equipment response, the brake becomes more lively and the driver becomes capable of braking with greater discrimination.

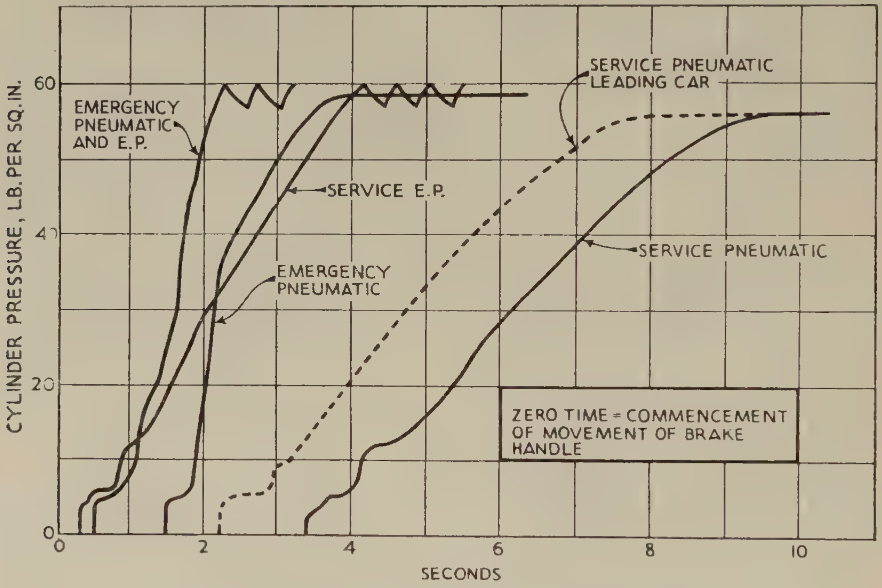


Fig. 5. — Stationary tests of rise of brake cylinder air pressure with different types of brake application; eighth car of an eight-car train.

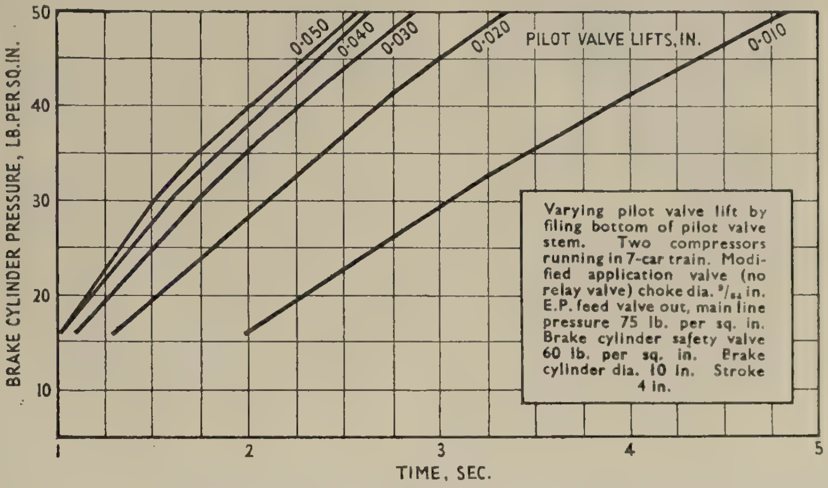
Apparatus.

Each part of brake apparatus suffers from inaccuracies in manufacture and maintenance. Examples are given below of the variations which result.

Brake cylinder air pressure. — Each wheel, if it is to be braked to the maximum, should have a torque applied to it proportional to the weight on rail. The rigging leverages and brake cylinder size are designed to this end and the term braking percentage is used to indicate the ratio of the force applied on the brake blocks on the two wheels of an axle to the weight-on-rail of that axle. It is obvious that with pneumatic braking the air pressure in the brake cylinder must be the correct value to produce the correct force, and be consistent at all times and all the way down the train. Unfortunately this is not so. When the driver makes an application, either pneumatically or electro-pneumatically, the pressures differ at the same moment in the brake cylinders.

From figure 5, it can be seen that with a pneumatic application there is a delay in the fall in train line (brake pipe) pressure which increases down the train, so that in the case shown the rise in pressure in the rear car began over 1 sec. later than the first; and after a period of 6 sec., while the pressure in the leading car was 44 lb. per sq. in., that in the trailing car was 30 lb. per sq. in.

If a controlled brake, working to the maximum possible use of adhesion, is in use, a means must be provided to halt the rise in pressure in the leading car before the wheels are picked-up and a skid has commenced. At this stage there are various lower air pressures in the cylinders in the other cars and the average resultant will produce a rate of braking lower than the maximum, to the degree that the effective average pressure is below the maximum for the adhesion.



Varying pilot valve lift by filing bottom of pilot valve stem. Two compressors running in 7-car train. Modified application valve (no relay valve) choke dia. 9/64 in. E.P. feed valve out, main line pressure 75 lb. per sq. in. Brake cylinder safety valve 60 lb. per sq. in. Brake cylinder dia. 10 in. Stroke 4 in.

Fig. 6. — Rise of brake cylinder air pressure, full E. P. application by D. B. V., with varying lifts of application valve pilot valve.

PART 2. — Characteristics in service of electro-pneumatic brakes; calculation of brake rigging efficiency.

In the previous part, the effect of air pressure rise in the leading car being more rapid than in the following vehicles, was discussed. With an electro-pneumatic brake, although the defect of a difference in time of first moment of rise of air in the brake cylinders is overcome, other troubles arise. The restrictions which can cause variations in rate of feed through triple valves appear with an E.P. brake in other places; in, for instance, the variation in lift of an application valve. Figure 6 shows such a variation, and a limit must be placed on this lift in manufacture and maintenance if it is not to be troublesome. Even so, the variation from the average reduces the possibility of maximum braking.

If E.P. feed valves are used, a variation

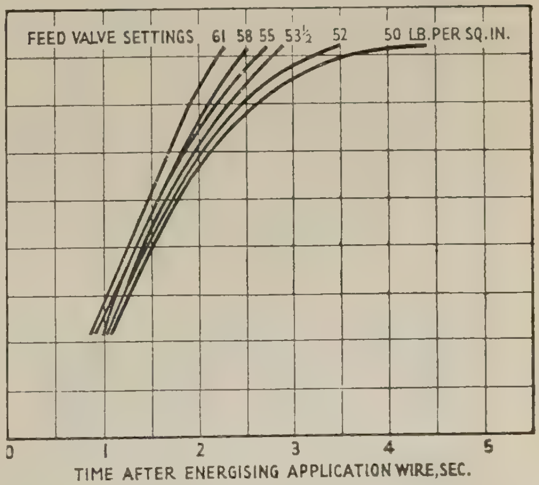


Fig. 7. — Rise of air pressure in 10-in. trailer car brake cylinder with various feed valve settings.

in setting can also produce variations in the rate of rise of brake cylinder air as can be seen in figure 7.

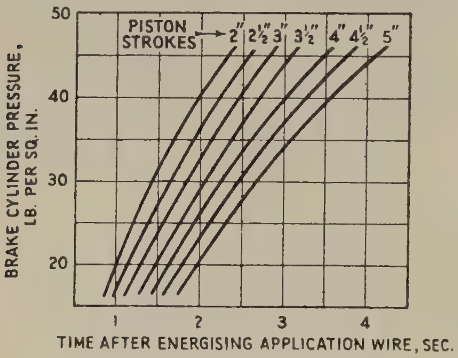


Fig. 8. — Rise of air pressure in 10-in. trailer car brake cylinder with various piston strokes.

Differences in brake cylinder stroke can also cause considerable variations in rates of rise as is shown in figure 8. An odd feature arises in this case, particularly where brake blocks are set below the horizontal centre line of the wheel and the ratio of loaded weight/tare is high. If all slack adjusters on a train are set to give the same cylinder stroke with a train empty, it is possible that, after a train has been in service and the slack adjuster has been working with the brake block in a lower position, due to passenger loading, the strokes will be reduced when the cars are once more empty of passengers. In a particular case where all cylinder strokes on a seven-car train were set accurately to 4 in. in the depot, on return from passenger service it was found that they had altered to 3 7/8 in., 3 1/2 in., 3 3/4 in., 3 7/8 in., 3 in., 3 1/8 in., and 3 7/8 in.

The differences in the effect of running in service were due to differences in the ratios of loaded weight/tare and the relative positions of the brake blocks on the wheels.

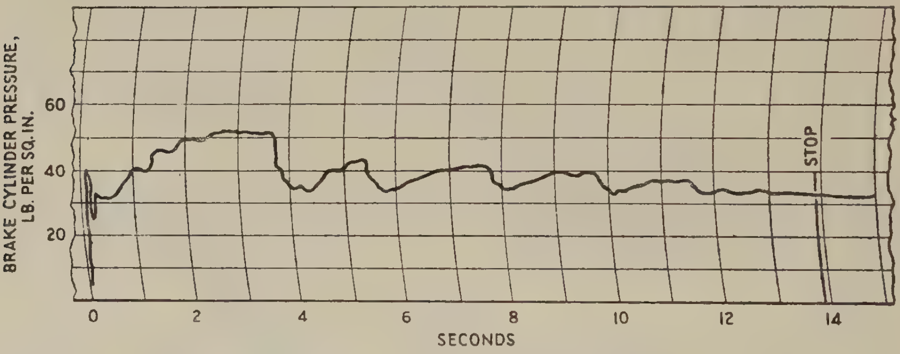


Fig. 9. — Retarder control of emergency brake stop from 41 m.p.h., leading car of empty 7-car 1938 tube stock train, non-metallic inserts, blowdown safety valve 31 lb. per sq. in., wheels and rails sprayed with water.

A further difference in brake cylinder pressure, from the maximum consistent with the adhesion, occurs with retarder control. The point in the cycle of braking where, it is judged, the limit of adhesion would be reached by the braking force becomes the point at which air must be vented from the brake cylinder. Due to the inertia of the air and controlling apparatus, the pressure in the brake cylinder is brought too far below the maximum, particularly in an emergency stop, and, with recovery, a succession of peaks and troughs occurs with an average which produces a braking rate lower than would be possible if the value could be held exactly on the limit of adhesion. Figure 9 shows a record of an emergency stop where the brake cylinder air has been controlled by a retarder, blow-down valve, and additional relay vent valve. It is possible that this form of control by relay valve is too crude and that the average may be brought closer to the maximum.

The considerations above emphasise the importance of air pressure measurement and the necessity of using an accurate master gauge for the setting of all devices rather than trusting to a local car gauge which is only an indicator.

Dynamic braking apparatus.

The various types of dynamic braking — regenerative, resistance, and eddy current — must all be scaled down by the degree to which the average is below the maximum. Variations in characteristic between machine and machine are appreciable, and resistors and other receivers of energy vary also within accepted manufacturing tolerances.

Brake rigging efficiency.

The term brake rigging efficiency is used to describe the ratio between the force on the brake block and the appropriate proportion of force of the brake cylinder piston multiplied by the leverage between piston and block. In designing leverages, cylinder sizes, and air pressures with pneumatic brake gear, it has been the practice in the past, where cast iron brake blocks are used, to calculate the percentage braking on motored axles at 85 % tare and free axles at 80 % tare. The figures obtained by this method were the values which would appear as denominators in the brake rigging efficiency ratio. The numerator varies with the type and layout of rigging. The efficiency may be as low as 66 % on an individual block on a car with a single brake cylinder, or as high as 95 % with individual brake cylinders for each block. These are values obtained on a stationary train with artificially vibrated rigging in the case of the car with single brake cylinder. There is scope for research in investigating the actual efficiency when the brake is doing work.

It will be appreciated that, if one axle has an average block rigging efficiency appreciably different from another, there is danger of pick-up on the more efficient axle, and allowance must be made therefore for the difference in the design of leverage or, with individual brake cylinder, in cylinder size and/or leverage.

PART 3. — Factors influencing coefficient of friction between block and wheel.

The factors now to be considered are the brake block itself, and the various conditions of control between wheel tyre and block.

Brake block.

In general the variation in brake block dimensions in manufacture can have but little effect on the block characteristics. Designed differences in size can however appreciably alter the temperature on a tyre surface during braking (*) and for that reason may alter its frictional characteristic. Once determined, however, it is probable that there is little difference between block and block on account of dimensional variation.

Variations in mix with non-metallic blocks may account for differences between block and block under test. The effect of scatter has not been analysed, but since each axle generally has four blocks, in practice the variations will tend to even out.

Block and wheel.

The variations in braking which occur through the mutual contact of wheel tyre and block are as follows :

Variation of the coefficient of fric-

(*) « The Measurement of the Temperature of Sliding Surfaces with particular reference to Railway Brake Blocks », by R. C. PARKER, Ph. D., B. Sc., and P. R. MARSHALL, Ph. D., B. Sc., Proc., Institution of Mechanical Engineers, Vol. 158 (1948).

tion (μ) of a brake block with speed. — Captain Galton's famous experiments in the last century established the fact that with a cast iron brake block acting on a steel tyre the coefficient of friction drops off with speed. The extent is considerable and figure 10 shows on the top curve the order of value of μ at the moment of application of the brake and, in the series of curves departing from it at various speeds, the form of curve which may be expected when a braking stop is made at various commencing speeds.

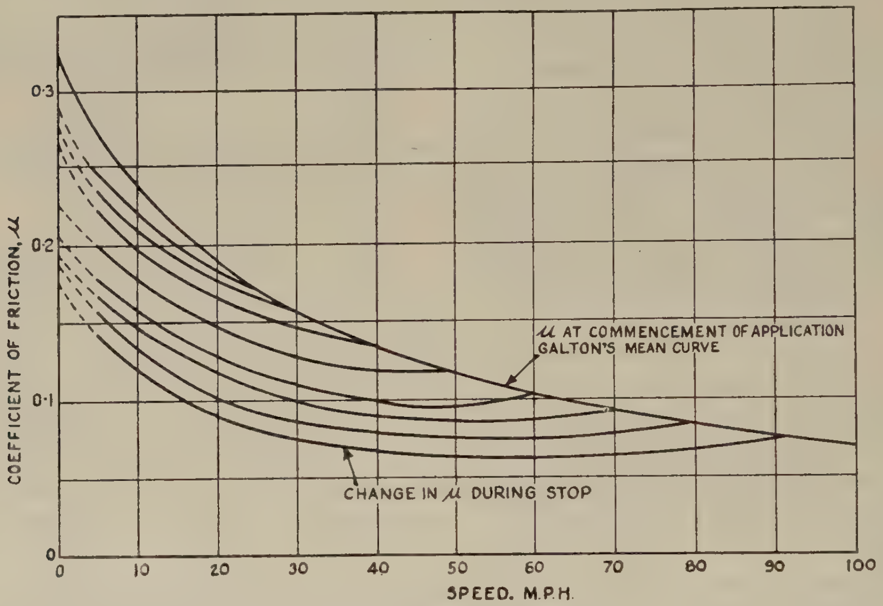


Fig. 10. — Change in coefficient of friction (μ) between block and wheel during stops from various speeds.

If the cylinder pressure is under no measured control, but can rise to, say, the equalising pressure in the case of a pneumatic application, the maximum value must be calculated on the basis of the maximum μ, in practice the coefficient at about 5 m.p.h. The braking force which can be obtained from the same brake cylinder pressure at a high speed will be reduced in the ratio of the coefficient of friction at the two speeds; or in the ratio of 0.10/0.27, or to 37%, at 60 m.p.h.

This factor of reduction in coefficient of friction with speed, when cast iron brake blocks are used, is probably the greatest deterrent to maximum braking with the majority of the stock in this country. It has a secondary effect in relieving the rate at which work is being done at the high speed, and thus reducing the demands on the tyres, but this takes place only at the expense of the long stopping distances, with their evident reactions on sections occupied, and timetables.

With non-metallic, composition, brake block materials the characteristic is flatter and research in this direction shows indications of finding a characteristic which can be controlled to a much greater degree than at present. It is possible that friction materials, or mixtures of various friction materials, with a flat characteristic will be found, but with an increase in cost of production. Against this higher cost can be set the saving in apparatus which is now used to compensate for the variation in coefficient of friction of the present materials.

The characteristic of a cast iron block, where the coefficient rises at the lower speeds, increases the tendency of a wheel to pick-up and lock, once slowing-up has begun on the rising part of the curve. Wooden brake blocks, as used on the Paris Métro urban sections, have a characteristic in which the coefficient drops off as the speed decreases. Should the wheel therefore slow-up because the braking force on an axle is too high for the adhesion, it can be observed that, at some point during the process, the wheel will speed up and again run at normal speed. This slowing-up followed by a recovery may take place several times during a stop. The same phenomenon has been observed with a non-metallic material, but not with such frequency during a stop.

Block μ and applied force. — A further factor which must be taken into account as a limiting variation is the change which can take place in the ratio of frictional force to applied force. With some types of non-metallic material the ratio decreases with increase in applied force. With a material of this type the reason may lie in a reduction in force per unit area with the rise in total force, but whatever the reason, if expressed as a coefficient of friction, a definite fall can be found. With controlled braking, where an increase in applied force is used to counter a fall in block friction with speed or for passenger loading, the effect may be important, particularly as the drop may differ between free or lightly loaded axles, and heavily loaded driving axles.

Block μ and tyre temperature. — With composition non-metallic materials very high temperatures may be reached and with somes types « fade » occurs. It is important therefore in brake design to know the characteristics of the braking material, whether metallic or non-metallic, when hot, if special requirements to cover for « fade » are to be avoided.

Block μ and weather. — If all braking were to take place in the dry, the problem of braking would be easier. Weather affects both the coefficient of rolling friction between wheel and rail and also the coefficient of friction between block and tyre. With cast iron the block μ is hardly, if at all, affected by wet, but with non-metallic materials μ rises with some and drops with others, while at the lower values of applied force the brake friction can fall off very rapidly. Since, however, we are concerned with maximum braking the lower values are not of immediate interest. The drop in friction at the higher values under wet conditions can, however, with some materials, be sufficient to call for considerable increase in applied force as compensation for drop in μ. If no extra pressure is available the rate of braking will fall.

If the fall in block μ could exactly equal a drop in adhesion under wet conditions, automatic compensation would take place, but it would be disconcerting for a driver to have two rates of braking, one for wet and one for dry conditions, with critical unstable conditions in transition. In practice, the settings for braking are designed for wet conditions and it is important therefore that the characteristic of the block should not change unduly between dry and wet. If there is to be a choice, it is better to have a slightly falling characteristic under wet conditions. A rising characteristic would tend to increase any tendency to pick-up.

Effect of wheel diameter with drum, disc and dynamic braking. — With torque braking where the frictional force is applied at the tread of the wheel, as in the usual form of block braking, any change in wheel diameter is automatically compensated for in the torque.

This effect can be seen from figure 11. It will be noted that the reaction from rail to wheel = 2 P and, assuming a constant coefficient of friction and applied force, the interacting circumferential force between wheel and rail remains constant with all wheel diameters and is limited only by ψ.

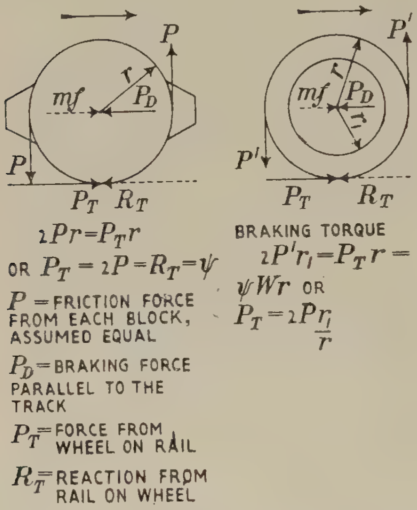

P
mf
P_D
r
P
P_T R_T
$2Pr = P_T r$
OR $P_T = 2P = R_T = \psi$
P = FRICTION FORCE FROM EACH BLOCK, ASSUMED EQUAL
P_D = BRAKING FORCE PARALLEL TO THE TRACK
P_T = FORCE FROM WHEEL ON RAIL
R_T = REACTION FROM RAIL ON WHEEL
P'
mf
P_D
r
r_1
P'
P_T R_T
BRAKING TORQUE
$2P'r_1 = P_T r = \psi W r$ OR
$P_T = \frac{2Pr_1}{r}$

Fig. 11 (left). — Constant interacting circumferential force between wheel and rail with block braking; and fig. 12 (right). Variation of force between wheel and rail with other forms of torque braking.

With other forms of torque braking, however, where the diameter of application remains practically constant, it will be seen from figure 12 that the force between wheel and rail varies as $\frac{r_1}{r}$ or with r_1 constant, as $\frac{1}{r}$. If pick-up is to be avoided it will be necessary to design on the basis of a fully-worn wheel with a resultant decrease, amounting in some cases to 10 % with a new wheel. The alternative is some form of adjustment of torque with change in wheel diameter, thus narrowing the percentage decrease.

It should be noted in the above that the friction force P is not the full force as a proportion has been used in retarding the rotating parts.

PART 4. — Conditions of adhesion between wheel and rail.

Wheel and rail.

The variation in conditions which may take place between wheel and rail is dominated by the adhesion between the two. The limitation which it imposes on braking has already been discussed. Many attempts have been made to arrive at a real value under varying conditions, but the scatter between points continues to be high. All that can be said is that the braking force can be no higher than the co-efficient of friction between wheel and rail will allow.

The other factors under this heading are as follows :

Variation in passenger loading. — With no compensating device for passenger loading, the maximum rate of braking decided by unloaded conditions may decrease by over 30 % with a heavily loaded train. If there is compensation, the effect on maximum braking will be controlled by the limits of the device.

With a load-compensating device applied to each car there will be variations between one end and the other which it would not be practicable to allow for. The load compensator can therefore only work to average car loading. It is also liable to variation in spring movement, and the mechanical limits to which it can be set.

With retarder control the total weight of the train controls the cylinder pressure, and it is therefore an average weight per car for the whole train which is measured. With variation in passenger loading between car and car, the braking must be set down to a value lower than would be possible if all cars were loaded in such a way as to give a proportionate increase in weight on all axles.

Weight transfer. — When a train is braked the forces acting on the various parts produce an effect which reduces the weight on some axles and increases it on others. The transfer, with intensive service rates of braking, can be over 10 %. A simple case is discussed in figure 13. For a fuller treatment reference may be made to an article in *The Railway Gazette* of June 16, 1950.

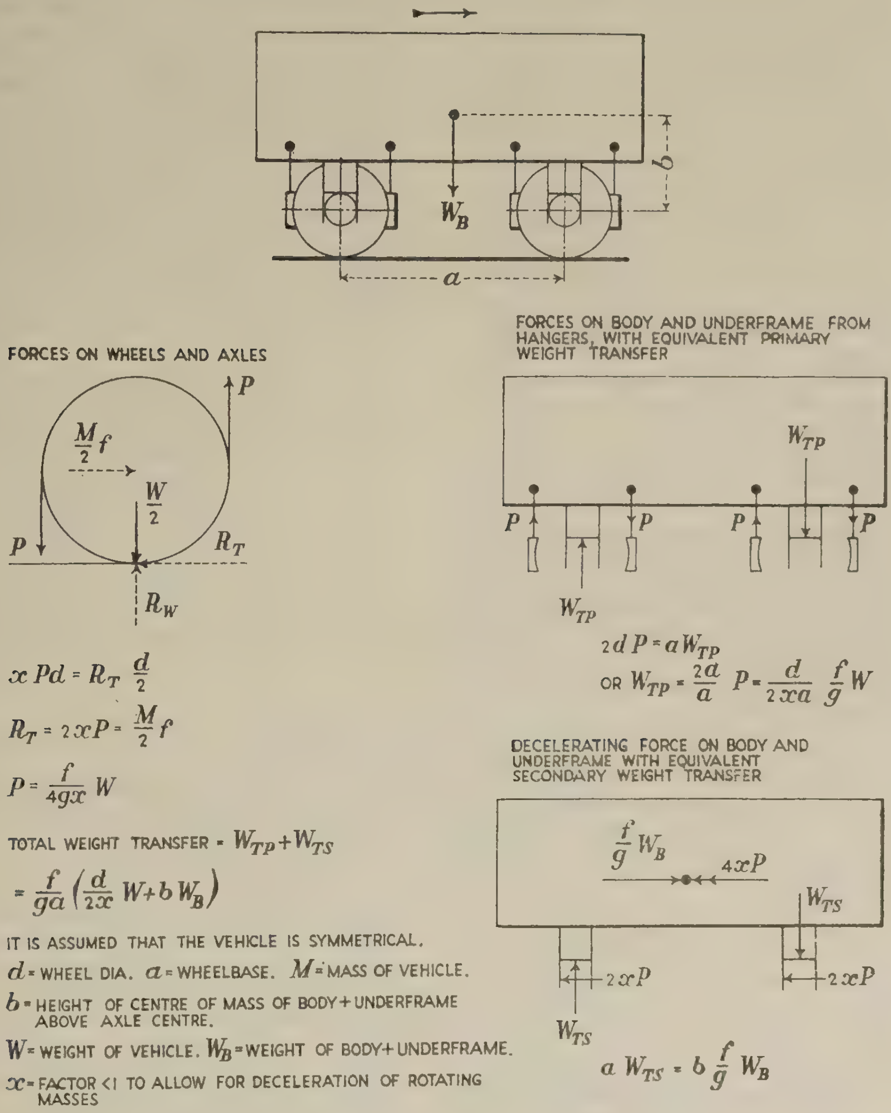

W_B
b
a
FORCES ON WHEELS AND AXLES
P
$\frac{M}{2}f$
$\frac{W}{2}$
P
R_T
R_W
$x\,Pd = R_T\,\frac{d}{2}$
$R_T = 2xP = \frac{M}{2}f$
$P = \frac{f}{4gx}\,W$
TOTAL WEIGHT TRANSFER = $W_{TP}+W_{TS}$
$= \frac{f}{ga}\left(\frac{d}{2x}\,W + b\,W_B\right)$
IT IS ASSUMED THAT THE VEHICLE IS SYMMETRICAL.
d = WHEEL DIA. a = WHEELBASE. M = MASS OF VEHICLE.
b = HEIGHT OF CENTRE OF MASS OF BODY + UNDERFRAME ABOVE AXLE CENTRE.
W = WEIGHT OF VEHICLE. W_B = WEIGHT OF BODY + UNDERFRAME.
x = FACTOR <1 TO ALLOW FOR DECELERATION OF ROTATING MASSES
FORCES ON BODY AND UNDERFRAME FROM HANGERS, WITH EQUIVALENT PRIMARY WEIGHT TRANSFER
W_{TP}
P
P
P
P
W_{TP}
$2d\,P = a\,W_{TP}$
OR $W_{TP} = \frac{2d}{a}\;P = \frac{d}{2xa}\;\frac{f}{g}W$
DECELERATING FORCE ON BODY AND UNDERFRAME WITH EQUIVALENT SECONDARY WEIGHT TRANSFER
$\frac{f}{g}W_B$
$4xP$
W_{TS}
$2xP$
$2xP$
W_{TS}
$a\,W_{TS} = b\,\frac{f}{g}\,W_B$

Fig. 13. — A simple demonstration of weight transfer.

Deposits on rail. — Under this heading can be included rusting, soot, oil and grease, used brake block material, dew, and rain. It is well known that a wet rail reduces adhesion, and that the decrease below the dry figure with a « greasy » rail may be even greater. The description « greasy » includes the condition of a rail unused during the night, with a layer of rust and a deposit of moisture. Fully wet conditions have been found to produce a reduction in coefficient between wheel and rail of the order of 20 %, which may be exceeded with the other forms of deposit.

With non-metallic blocks it has been found that the products of braking contaminate both rail and wheel. The combination of greater smoothness of wheel tread with the contamination reduces the coefficient of friction between wheel and rail by about 20 %. The degree to which each of the two factors contribute to this reduction is not known.

Effect of speed. — Various opinions on the reduction which is said to occur with higher train speeds were quoted in a paper read before the Institution of Locomotive Engineers in 1948 (*). Whether the effect described is produced by the peripheral speed of the wheel or by some other factor in wheel/rail contact which is a result of that speed, and varies, say, with track conditions and out-of-balance of rotating parts, is not known.

Data regarding the effect of speed on adhesion are derived mainly from tests of tractive effort. It would appear probable that braking conditions differ sufficiently to make it necessary to approach the problem, initially at least, as a distinct issue.

Mutual wheel/rail contact. — In mentioning the effect of speed, reference was made to other factors affected by speed. These include the condition of joints, crossovers, points, and worn rails. The supporting ballast sleepers and chairs must also be considered as contributory agents in affecting the contact between wheel and rail. Little information is available to enable a designer to allow for such factors.

Gradients. — The rate of maximum braking on a gradient is automatically increased or decreased by a gradient.

On railways using steel rails and wheels, the effect of gradient on adhesion is negligible. For instance, a 1 in 30 grade alters the weight available for adhesion by only 0.66 %. On the other hand the force assisting or reducing braking is appreciable (fig. 14). If maximum use is being made of the adhesive weight, the maximum braking force which can be exerted by torque braking = ψ W cos α.

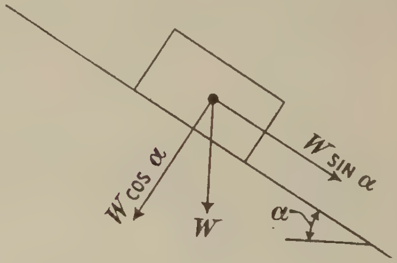


Fig. 14. — Force assisting or reducing braking on a gradient.

The net maximum braking force = W (ψ cos α $\pm$ sin α).

With a 1 in 30 grade and an adhesion ratio of 0.2 the net maximum braking force = W (0.20 $\pm$ 0.033) :

— this being an alteration of 16.7 %.

It is to be observed that, where an equalised brake cylinder pressure produces a braking force on the limit of adhesion on the level, the same cylinder pressure would tend to cause pick-up on a down grade, when braking with unbraked wagons, if the equalised cylinder pressure were on the limit for the locomotive operating on its own. This is due to the

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(*) « Adhesion and Friction in Rail Traction », by J. KOFFMANN, read at the Institution of Locomotive Engineers, October 20, 1948.

reduction in torque absorbed in such cases by the rotating parts and the consequent higher net force at the wheel tread.

Train.

Frictional resistance. — Frictional train resistance helps braking and thereby always reduces the work to be done by other means of braking. In the case of pneumatic tyres this resistance is relatively high and the contribution to braking is appreciable. With steel wheels the amount is small and may generally be neglected.

Windage. — On the other hand the effect of windage is appreciable whether self-made or not.

On occasions when winds are high and follow a train, extra braking force would be required, and should the braking be on the limit, the extra force might cause wheel pick-up.

PART 5. — The question examined from the angle of economics.

Economic factors.

Any change in a braking system involving extra cost must be judged on the advantages to be gained in greater safety, improvement in train handling, reduction in overlaps, or easier maintenance. The factors examined below indicate the difficulties.

Range of air pressure. — It was explained in Part 3 that, if maximum braking were required on a train fitted with cast iron brake blocks from speeds of the order of 60 m.p.h., the force available for application to the brake block must be of the order of 2.7 times that needed for the speed at the limiting value of coefficient of friction. If the ratio of loaded weight to tare is, say, 1.3, the total allowance for increase in air pressure above the minimum will be of the order of 3 1/2 times the minimum.

If the brake cylinder blown-down safety valve were set at, say, 30 lb. per sq. in., a pressure of 105 lb. per sq. in. in the brake cylinder would be required at 60 m.p.h. Taking the brake cylinder safety valve at 5 lb. per sq. in. above, the brake pipe feed valve would be set at 120 lb. per sq. in. and the minimum main reservoir pressure at 130 lb. per sq. in. These are high pressures for a pneumatic system, and special measures would be required in the design.

Extra expense may be justified because the higher rate of braking might reduce the trains required for a given service. If applied to emergency braking, overlaps could be reduced and an increase of service could be run for the same maximum train speeds. Each case would, however, require justification on its own merits.

Other solutions, such as the introduction of an extra cylinder operating only above certain speeds, might be examined, but again the cost rises.

Effect on wheel and tyre of high rates of works. — The rate of doing work expressed as HP is given by the equation :

HP = 0.272 WVf where W = weight (tons), V = speed (m.p.h.), f = deceleration (m.p.h./sec.).

With allowance for rotating parts the values would be about 10 % higher.

At 50 m.p.h., therefore, with an axleload of 10 tons and with a rate of braking of 2.65 m.p.h. per sec., the HP per axle is 400. With two brake blocks per wheel, the HP per block is 100, a high rate of doing work for such a small mass. The surface of the tyre under the block and the block itself are subject to high temperatures. With non-metallic blocks values of the order of 1 500° C. have been estimated (*) for tyre face temperatures with a kinetic energy of 6×10^5 ft. lb. per block at 60 m.p.h. and an applied force on the block of 3 560 lb.

(*) « The Measurement of the Temperature of Sliding Surfaces, with particular reference to Railway Brake Blocks », by R. C. PARKER, Ph. D., B. Sc., and P. R. MARSHALL, Ph. D., B. Sc., Proc. Institution of Mechanical Engineers, Vol. 158 (1948).

The tyre suffers from the generation of high temperatures by a change in structure and hot spots resulting in the production of martensite may end in thermal checking and metal plucking.

For high speeds it may be necessary to consider some form of friction braking, *i.e.* disc or drum, in which the heat developed may be controlled under conditions which can be designed.

If the full rate of braking is to be worked to at the higher speeds, a mixture of friction braking and dynamic braking may be required, but both this mixture, and the drum or disc braking, lead to greater expense than the simple arrangement of block on wheel tread. They must therefore be considered in this light.

Stresses. — With the greater forces required to produce the higher rates of braking at the higher speeds, the stresses in the bogie structure and rigging rise to values above those normally dealt with. In consequence points of weakness may arise which would be concealed by allowances made for other reasons.

The stresses in axles and wheels will also be increased and must be considered as a possible cause for increase in dimensions.

Rate of wear of blocks and tyres. — With steel wheels, although the amount of work which is done in braking from a given speed is substantially the same whatever the rate of braking, it does not follow that the wear of blocks and tyres will remain the same. The higher temperatures which result from the higher rate of doing work may alter the rate of wear, and the higher cost must be included in any overall estimate of the benefits to be derived from shorter braking distances and times.

Standardisation of parts. — Mention has been made already of the fact that for maximum braking each axle must have a force applied to it proportional to the limiting tangential force between wheel and rail. This is taken to be proportional to the axle weight. It would be necessary, at least with most multiple-unit stock, to have a variety of brake cylinders and, possibly, lever drillings. In practice, the number is reduced to as few standard sizes and dimensions as are possible without great divergency from the ideal. To the degree however that there is a departure, there is a loss from maximum braking.

Space. — Even if all requirements for maximum braking can be fulfilled there may be occasions, as for instance on tube stock, where the limitations of space preclude the fitting of the required extra gear. The difficulty, for instance, in obtaining a straight run for the train line brake pipe is an example of the restrictions which may occur.

It will be gathered from the comments made on the various limitations that knowledge on restrictions to maximum braking is by no means complete.

The author wishes to thank Mr. A. W. MANSER, Chief Mechanical Engineer (Railways) and the London Transport Executive for permission to publish the information contained in this article.

[625 .112 (94)]

The standardization of railway gauges in Australia,

by W. D. Chapman, M. C. E., D. Eng.

*(Member)**

(From *The Journal of the Institution of Engineers, Australia,* for July-August 1952).

Summary. — The paper is a review, from an engineering and economic viewpoint, of the problem of re-gauging certain Australian Railways.

History in brief.

It is not intended to trace in full the history of the five main gauges of Australian Railways, nor the subsequent investigations, except inasmuch as to refer to some important dates and reports.

The first noteworthy point is that several years before the first railway was constructed in Australia, Great Britain followed the recommendation of a Parliamentary Commission and in 1846 adopted 4 ft. 8 1/2 in. as the standard railway gauge.

In 1897, before Federation, a conference of Railway Commissioners of Victoria, South Australia and New South Wales, recommended the adoption of the 4-ft. 8 1/2-in. gauge, conversion then being estimated to cost £2 360 000 and to take five years.

Many other reports followed after Federation, as a result of which some seven Commonwealth Acts of Parliament re-affirmed the principle of the 4-ft. 8 1/2-in. Standard Gauge.

These are as follows :—

1911. *Kalgoorlie to Port Augusta Railway Act* (4-ft. 8 1/2-in. gauge specified).

1913. *Pine Creek to Katherine River Railway Act* (3-ft. 6-in. gauge, but provision in permanent way and works for subsequent alteration to not less than 4-ft. 8 1/2-in. Roadbed, structures and sleepers were provided for 4-ft. 8 1/2-in. gauge).

1924. *Crafton to South Brisbane Railway Act* (« Standard Gauge » specified).

1926. *Railways (South Australia) Agreement Act* (Line to Alice Springs alternatively 3-ft. 6-in. extension from Oodnadatta, or 4-ft. 8 1/2-in. from Tarcoola. The former was adopted.

1935. *Port Augusta to Port Pirie Railway Act* (4-ft. 8 1/2-in. gauge).

1946. *Railway Standardization Act No. 50.*

1949. *Railway Standardization (South Australia) Act No. 83.*

An interval of eleven years will be noted, during which no legislative action was taken, viz., from 1935 to 1946.

It was during this interval and for some years previously that there was a measure of uncertainty as to the future role of railways. The rapid development of road and air transport and the state of the railway finances when £62 000 000 deficits were piled up in the ten years before the war, were contributory to this viewpoint.

Then came World War II, bringing an average increase in traffic over all systems of 25 per cent. for the six years, with a peak of 50 per cent. increase in 1942/43. The railways, although starved as to capital expenditure for over a decade, coped with this extra load in general with the aid of some relief by restrictions on civilian traffic amounting to 20 per cent. or more, and a capital expenditure of £21 000 000. There were, however, certain sections or systems on which the overload was four to five times the normal, for example, the East-West and North-South Commonwealth-owned systems where the disadvantages of the non-standard gauges were very apparent.

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* This paper, No. 1057, was presented before the 1951 Engineering Conference, in Brisbane, on 3rd April, 1951.

The Author is Director of Civil Engineering, Railway Standardization Division, Department of Fuel, Shipping and Transport, Commonwealth of Australia.

In 1945, following the « lessons learned » during the war, a new appraisal was in the Report by Sir Harold Clapp, K.B.E. — « The Standardization of Australia's Railway Gauges, » March, 1945.

In addition to the gauge question, the objectives of modernization of equipment and standardization of rolling stock, operation, and rating, were covered in this report and the organization necessary to implement the recommendations was outlined.

The « Clapp » Plan recommended a first phase, Proposals *A.*, *B.*, and *C.*, estimated to cost £76 751 000 and to take eleven years to complete.

The time element is the matter of which, in the main, there has been a lack of appreciation, for it has an important bearing as to whether the work is within the capacity of the Nation's resources.

The first phase included an independent standard gauge line Fremantle-Kalgoorlie, the conversion of the South Australian and Victorian Systems, and certain lines in Queensland linking Townsville with the South and to Darwin.

The later phase, Proposals *D* and *E*, the conversion in Western Australia and Queensland, was recommended for further consideration, the estimated cost being £133 000 000. Tasmania was not included nor the North-South Line, except the portion to connect with Townsville already included in the first phase.

Following upon the Cabinet acceptance of the « Clapp » Report, discussions took place in 1946 between the Commonwealth and the States with a view to establishing a basis for agreement.

These negotiations broke down. All States objected to the proposals as to the financial allocation suggested, viz., Commonwealth one fifth, States four fifths, and this was amended to 50/50. All States, excepting Tasmania, desired practically all lines regauged, the total estimated cost, including associated works, thus being increased to £286 600 000. South Australia asked that the existing agreement to complete the North-South Line should be honoured in place of, or in addition to, the Townsville-Darwin connection. Queensland objected to the inland Bourke-Hughenden route.

Negotiations were continued with the three States of New South Wales, Victoria and South Australia, and agreement was reached with the Premiers of these States.

The Commonwealth Act No. 50 of 1946 received Assent on 15th August, 1946, the South Australian Act No. 31 on 10th December, 1946, and the Victorian Act No. 5343 on 21st December, 1948. The Government of New South Wales did not ratify the Agreement.

The next step was a separate agreement with South Australia following which the Commonwealth Act No. 83 of 1949 was Assented to on 28th October, 1949, and the South Australian Act No. 49 of 1949 Assented to on 12th December, 1949.

This Act provides for the conversion of the South Australian 5-ft. 3-in. system and the 3-ft. 6-in. gauge lines of the South Eastern and Peterborough Division. The Commonwealth undertakes the conversion of the 3-ft. 6-in. lines of the North-South Railway and the construction of the link from Alice Springs to Birdum and also to « take reasonable steps to ensure the acquisition of the Silverton Tranmway » and to hand this link over to South Australia for conversion. With the works included in these Acts completed, a major stage towards the completion of the scheme would be reached in that the Commonwealth, New

South Wales and South Australian Systems become a block of standard gauge lines.

The continuity between South Australia and Victoria would, however, be broken and the discontinuity between New South Wales and Victoria would remain. The Queensland and Western Australian systems are still not included and require further examination.

It is proposed therefore, to follow with an appreciation, mainly from an engineering and economic point of view, of the problem or re-gauging as presented at the present time, over five years since the Report of March, 1945.

The objectives.

The objectives in general, including re-gauging are :—

(1) That the deferred maintenance, renewal and modernization programmes of the railway systems should be speedily carried out.

(2) That provision should be made for the future requirements for traffic increase.

(3) That re-gauging to 4 ft. 8 1/2 in. should be advanced as far as possible in conjunction with Objectives 1 and 2 which are now urgent matters.

The considerations.

(1) *The essentiality of railways :*

The main consideration for the expenditure involved in objectives 1 and 2 is that railways are now, and will still be in the future, essential in the Nation's economy.

The Railway Capital Expenditure programme, in hand or projected, as far as can be ascertained is in the neighbourhood of £400 000 000 for a five- to ten-year programme. This may seem an inordinate expenditure in view of the booked capital value of railways in 1947/48, viz., £338 650 398.

On the other hand, by comparison, the capital value of new road transport vehicles in the year 1949/50 was over £100 000.000.

In viewing the internal transport costs and the task, the railways share can be seen in better perspective. In the latest assessment available, viz., 1947/48, the total expenditure was as follows :—

	£	%
Marine (Coastal only) . .	20 000 000	3.6
Road Transport	419 000 000	76.2
Railways	85 000 000	15.5
Tramways	15 000 000	2.7
Air	11 000 000	2.0
	£550 000 000	100.0

or approximately 30 per cent. of the estimated total national expenditure of £1 870 000 000.

This high proportionate cost as compared with many other countries is obviously due to the low average population density with concentrations in certain areas, fairly widely separated.

The estimated transport task for the same year is shown in Table I.

Table I.

	Passenger Traffic.			Freight Traffic.		
	Passenger journeys.	Passenger miles.	%	Tonnage.	Ton-miles.	%
Marine (Coastal) . . .	105 000	52 000 000	0.2	8 025 000	6 620 000 000	37.7
Road	Not. available	Upwards of 14 500 000 000	56.7	140 000 000	5 600 000 000	31.9
Rail	510 767 000	7 038 000 000	27.5	39 946 000	5 317 000 000	30.3
Tramways . . .	1 189 000 000	3 481 000 000	13.6	—	—	
Air	1 208 000	503 000 000	2.0	27 000	13 000 000	0.1
Total :		Upwards of 25 574 000 000	100.0		17 550 000 000	100 0

Thus, railways carried 30.3 per cent. of the freight traffic and 27.5 per cent. of the passenger traffic for 15.5 per cent. of the total cost.

Turning to the detailed costs, and for simplicity again, including all governmental and user espenditure, these averaged as set out in Table II.

Table II.

	Total average cost per passenger-mile. (Pence)	Total average cost per ton-mile freight. (Pence)	
	1947/48	1947/48	Future target.
Sea	2.25 to 2.75	0.81	0.70
Road . . .	1.0 to 4.0	13.7	11.0
Railways .	0.83*	2.19	1.6
Tramways .	1.07		
Air	3.43	41.74	32.0

* With proper costing this would probably be actually much higher and the freight costs relatively lower.

It will be noted that speed has to be paid for and also that the disparity in cost is less in the case of passenger traffic than in freight.

Speed and frequency have a great value in passenger traffic. The saving of a day for a passenger may amount to pounds. There is also a social value which cannot be assessed. On the other hand, the saving in a day's transport on a machine worth, say, several thousand pounds is only a few shillings in interest. On most freight it amounts to a fraction of a penny. The main consideration for possibly 90 per cent. of freight traffic is cost.

This is not to say that certain passenger traffic should not be carried by rail. This depends on the density of traffic, and distance, and the speed and frequency which can be provided.

In this study, however, passenger traffic will be regarded as of less importance than freight traffic and not likely to increase at the same rate.

There is already a back lag to be overtaken in that it is estimated that 1 700 000 000 ton-miles could not be handled by rail last year, and this is estimated to have cost the Nation in the neighbourhood of £20 000 000.

Looking ahead, in view of immigration proposals, decentralization, development and increased production per head of population, the present railway freight task should increase by at least 30 per cent. every ten years. Thus, in ten years the railway freight task may well be of the order of £9 000 000 000 ton-miles.

With a possible future target of 1.6d. per ton-mile by rail and for heavy road transport, including the cost of roads, of not less than 3.5d. for the 132-mile average haul the savings to be effected by keeping the proper proportion of the traffic on rail are considerable.

If railways are rehabilitated, the total annual cost of freight haulage should be of the order of £60 000 000, for the 9 000 000 000 ton-miles.

If, on the other hand, railways are not rehabilitated and can carry only the present tonnage, the total cost would be at least £102 000 000. Thus, the additional cost would be of the order of £42 000 000.

These figures are on the 1947/8 basis as to costs. There may be rises in costs in the future if the present trends continue, but in that case the disparity would be still greater.

Therefore, there is no doubt as to the economics of railway rehabilitation and modernization expenditure, providing it is judiciously expended and that railways are used on their proper task, nor is there any doubt that railways will be essential for a long period ahead.

(2) *The considerations for re-gauging to 4 ft. 8 1/2 in.:*

In the first place it may be as well to set out the disadvantage of the diversity of gauges and then to determine as far as possible the savings under two headings, viz., those which are assessable and those which are not readily assessable, and in any event only progressive with the scheme.

The disadvantages are as follows :—

1. The universal use of locomotives and rolling stock is not possible, therefore there can be no common pool from which to meet peak requirements. Each system has to maintain its own reserve, except in the case of South Australia and Victoria where some interchange is possible.

2. Extensive sidings and equpment must be provided and there is also dual provision in such matters as locomotive running sheds and the like.

3. The cost of transfer and the waste of manpower at break of gauge stations is unavoidable although it can be reduced for certain types of trafic by suitable equipment.

4. There are delays to locomotives and rolling stock, and the employment of locomotives for shunting is occasioned.

5. Theft, deterioration of perishables, damage, delay and other losses may be occasioned.

6. Mixed consignments are difficult to handle.

7. Load capacity and internal dimensions of freight waggons vary on the various systems, necessitating splitting of consignments and light loading in some cases.

There are, in addition, certain other disadvantages from a defence point of view which will not be enumerated in this paper.

As well as the disadvantages of the diversity of gauges, there are certain inherent disadvantages of the narrow gauge systems which are an important consideration both for passenger and freight work. High speeds can never be attained for passenger traffic even with a far higher standard of maintenance to ensure safety and comfort. The limited width of vehicles is a restriction on comfort and convenience. For freight traffic the disadvantages of the 3-ft. 6-in. gauge have more an economic value.

Over a certain traffic density any saving in capital cost of the narrow gauge is more than offset by extra cost of operation. This extra cost is occasioned by smaller average train loads and lower load/tare ratios. The lower average speeds and higher cost of maintenance for a given speed are also factors as in the case of passenger work.

In one case studied recently, for a proposed new line, the density at which it was estimated that the 4-ft. 8 1/2-in. gauge would be as advantageous as the 3-ft. 6-in. gauge was 125 000 tons. For any tonnage over this amount the 4-ft. 8 1/2-in. gauge becomes progressively more advantageous.

On an average, from a recent study, the saving in cost per net ton-mile for locomotive power alone on 4-ft. 8 1/2-in. as compared with 3-ft. 6-in. gauge can be taken as 0.1 pence per ton mile.

As to the total cost of re-gauging, it is not at this stage possible to give more than an approximate estimate. The cost would depend on—

(i) the extent of the conversion,

(ii) the rate and the order of conversion.

(*i*) *The extent of the conversion :*

The programme should include all main lines, connections and certain branch lines.

On most of the main lines and many branch lines on the 3-ft. 6-in. there is an assessable saving in operation and transfer costs by conversion, but it would be necessary to include some proportion of the lines with lighter traffic density which are necessary to leave a workable system.

For the 5-ft. 3-in. gauge lines there is no direct saving in operation except costs due to break of gauge, but there are the unassessable savings which would progressively increase with the progress of the scheme.

A note as to the economics of branch lines on both the 3-ft. 6-in. and 5-ft. 3-in. gauges should be made at this stage.

These branch lines, many of which were developmental and date from pre-motor transport days, should be reviewed from time to time as to whether re-gauging, retention on the existing gauges, or closing, is justified.

All costs should be considered, not merely user costs.

As to re-gauging, the annual costs of transfer and the extra cost of operation as an isolated line must be considered and balanced against the annual cost of re-gauging. Volume of traffic may not be the deciding factor. Length of line and cost of conversion are also factors.

A recent study showed that a minimum annual estimated tonnage per mile of line of approximately 1 100 tons in 1969/70 was required to justify present re-gauging of 3-ft. 6-in. gauge branch lines to standard gauge. This figure is equivalent to a present traffic volume of approximately 786 tons per mile of line. This study assumed an increase in volume of traffic of 40 per cent. over present figures in 20 years, and an increase in charges of 25 per cent.

Thus, 78 600 tons of traffic per annum would be required to justify conversion of a line 100 miles long, and only 7 860 tons per annum for a line 10 miles long.

The considerations in the closing of a line were enunciated in 1937*, the determination recommended being from the community point of view. On this basis many-so-called non-paying lines should remain.

If not re-gauged, their closing may be accelerated in some cases by the added costs of working as an isolated branch line. Two intermediate stages present themselves before final closing, viz.—

(*a*) closing to passenger and parcels traffic,
(*b*) closing to all but seasonal traffic.

Another matter affecting the cost of re-gauging should also be considered, viz., the co-ordination of rail and road transport through focal stations. This is an important matter for general consideration by Tranport authorities but should receive close attention when re-gauging of any line becomes imminent.

(*ii*) *The rate and order of conversion :*

The ultimate cost of re-gauging would be reduced by carrying out as much work as possible in conjunction with normal rehabilitation and will be influenced also by the order in which the work is carried out.

The ultimate benefits to be derived will be attained earlier the faster the rate of re-gauging.

On both these counts, the narrow gauge has the higher priority than the 5-ft. 3-in. gauge, although from other major considerations the re-gauging of the large block of 5-ft. 3-in, gauge lines in Victoria and South Australia is desirable. The proportion of work that can be ultimately of value on the 4-ft. 8 1/2-in. gauge is less in the 3-ft. 6-in. gauge and the immediate assessable savings are greater, although this is offset to some degree in that early conversion of the 5-ft. 3-in. gauge would accelerate the realization of the unassessable savings.

(3) *The considerations in combining re-gauging with rehabilitation :*

Much urgent rehabilitation must go on before re-gauging can be done, but apart from the importance of the rate of progress on ultimate cost and ultimate benefit, an early determination of policy will give an ultimate saving in cost.

On the 3-ft. 6-in. gauge, when earthworks require attention banks can be rebuilt with the slight extra width, new or replacement bridges can be rebuilt suitable for 4-ft. 8 1/2-in. and extra clearances provided often at very little extra cost. Sleepers for the 4-ft. 8 1/2-in. gauge could be used in re-sleepering operations if conversion is programmed before the life of the sleeper is likely to expire. The extra cost of the longer sleeper is easily determinable.

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* Report of the Royal Commission on Transport, Queensland 1937, pp. 13-17 and Appendix E.

Some of the 3-ft. 6-in. gauge rolling stock already in running or under purchase will be convertible but certain stock, such as locomotives, should be types and designs suitable for disposal to other 3-ft. 6-in. systems when no longer required, as conversion is difficult if not impracticable.

The extensive programme on the 5-ft. 3-in. gauge presents no problem as to locomotives and rolling stock which can and are being designed for future conversion at a low cost. On these systems earthworks and clearances are no problem. New track work which, although few new lines are under construction, is being rapidly added to by duplication and extension of sidings does, however, add to the cost of conversion.

(4) *Future administration and organization :*

The last consideration to be mentioned is largely outside the province of this paper but it is included as of fundamental importance in this problem. Heavy capital expenditure on railways is justified only if railways are allotted their proper proportion of the Nation's transport task.

The steady development towards a national « transport policy » over the past quarter century, and the prospect of its continuance, is an assurance in this regard.

The programme.

In view of the foregoing it will be appreciated that there is no definite programme to submit at this stage. Furthermore, the question of re-gauging in Queensland has not been studied for some time.

In order, however, to gain some conclusion as to the economic merit of re-gauging it is submitted that from the engineering point of view it could be reasonably practicable to carry out in 20 years the following :—

Commonwealth Railways :
- Conversion of Central Australian and Northern Australian Railways.
- Construction of the line Alice Springs-Birdum.

South Australia :
- Conversion of 3-ft. 6-in. lines excepting Port Lincoln Division.
- Conversion of 5-ft. 3-in. lines.

Victoria :
- Conversion of 5-ft. 3-in. lines excepting certain branch lines.

Western Australia :
- Conversion of approximately half the system.

New South Wales :
- Conversion of the Silverton Tramway.

Queensland :
- This system needs further consideration, but possibly 3 000 route miles of the total should be included. Victoria, Queensland, and Western Australia conversion would need further study as the scheme progressed.

The economics of re-gauging.

The cost of this programme would average £4 000 000 per year and total £80 000 000 in 20 years.

The return on this expenditure at the end of 20 years may be considered in three sections as follows :—

(1) *Saving in working expenses on re-gauged 3-ft. 6-in. lines,* including transfer costs on remaining branch lines		£1 179 000
(2) *Saving due to uniform gauge :*		
(*a*) Reduction in staff and establishments at break-of-gauge points	£211 000	
(*b*) Elimination of actual transfert costs	£ 98 000	
(*c*) Release and improved utilization of locomotive and rolling stock	£500 000	
		£ 809 000

(3) *Unassessable savings :*

Saving in cost of manufacture, store holding, etc., with standard equipment;

Saving due to the establishment of a pool of rolling stock;

Savings due to faster transit of passengers and goods and readier handling of special peak load such as livestock in time of drought, etc.

Total more than : £1 988 000

As the annual cost at an interest rate or 3.25 per cent. and redemption at 0.50 per cent. is £3 000 000, there is £1 000 000 per annum to be paid for the unassessable advantages enumerated under Item (3) above. Before and beyond the 20-year period, the annual costs and annual savings vary.

The annual cost rises to a peak and then disappears with redemption. The annual savings steadily increase with increase in traffic.

Conclusions.

From an engineering point of view, it is desirable to establish a definite railway policy and programme for, say, 20 years.

The policy should include the following :—

(1) Railways rehabilitation and modernization, including certain new works and electrification, as justified by present and estimated future transport requirements.

(2) That this work should be done either in conjunction with re-gauging, or as far as possible adaptable to standard gauge.

(3) The programme should include all main lines, connections and branch lines as are necessary to leave the railway systems workable, but certain outlying systems, developmental branch lines, etc., should be reviewed from time to time as to whether re-gauging, retention of the existing gauge, or closing, is justified.

(4) The programme should be elastic and accelerated or retarded as circumstances arise, the main factors being shortage or surplus of labour and materials available.

(5) During the early years, the problem of special mechanical equipment for widening banks and cuttings, trackwork in re-laying and re-gauging operations should be attacked. Concurrently, the selection and training of personnel should be undertaken.

From an economic point of view the loss is unlikely to be more than £1 000 000 per year, but in all probability the scheme would show a considerable saving over interest and redemption charges if the unassessable advantages are realized.

[656 .254 (73)

Trains run in either direction on both tracks,

with new ACL installation of C.T.C.

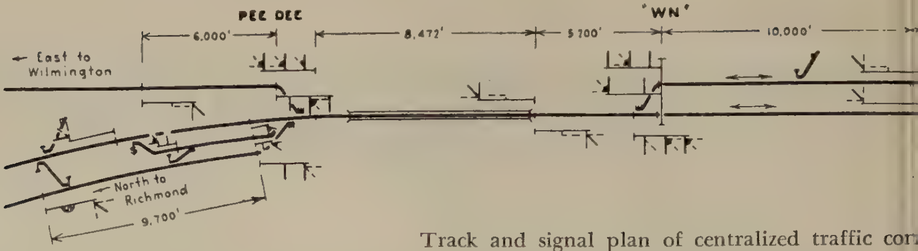
(Railway Age, July, 21, 1952.*)*

The C. T. C. begins at FY, at the new north end of the Florence yard.

To increase track capacity and improve flexibility of train operations, the Atlantic Coast Line has installed centralized traffic control on 2.5 miles of single track and 10 miles of double track between Florence, S. C., and Pee Dee. Florence is 298 miles south of Richmond, Va., on the double-track main line to Jacksonville. From Florence, a secondary main line extends west to Augusta, Ga., where a connection is made with the affiliated Georgia to Atlanta.

Florence is a division point, and has a large yard. From the passenger station, at the south end of this yard, double track extends north 10 miles to « WN » (see diagram), beyond which there is single track 2.5 miles on a trestle and a bridge over the Pee Dee river to Pee Dee. From Pee Dee, the double-track main line extends north toward Richmond, and a single-track main line extends east to Wilmington, N. C.

Previous to the installation of C. T. C.,

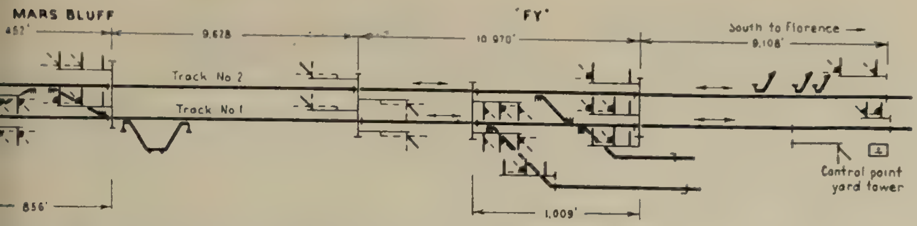


Track and signal plan of centralized traffic con

a 16-lever electropneumatic interlocking machine at Pee Dee controlled three switches and five signals, as well as the remote control of the end-of-double-track switch and three signals at « WN » near Mile Post 283. Also, there was a 16-lever electromechanical interlocking machine at « FY », at the north end of the old Florence yard. This machine controlled five switches, two derails and eight signals. All signals were of the semaphore type. The old signalling through this territory was arranged for righthand running only on the double track.

Southbound passenger train passing northward home signals at the Mars Bluff crossover.

At the time C. T. C. was installed, the Florence freight yard was modernized and extended northward. This required moving the « FY » interlocking so as to control the north end of the new yard. Also, a new double crossover layout was required at Mars Bluff, approximately midway between « FY » and « WN », to provide the desired flexibility of operation. The new C. T. C. system, with the control machine in the yard office at Florence, now controls the power switches and signals at « FY », Mars Bluff, « WN », and Pee Dee, and the signals at the north end of



ntic Coast Line between Pee Dee, S. C., and Florence.

the Pee Dee siding. The signals function for train operation in either direction on both main tracks.

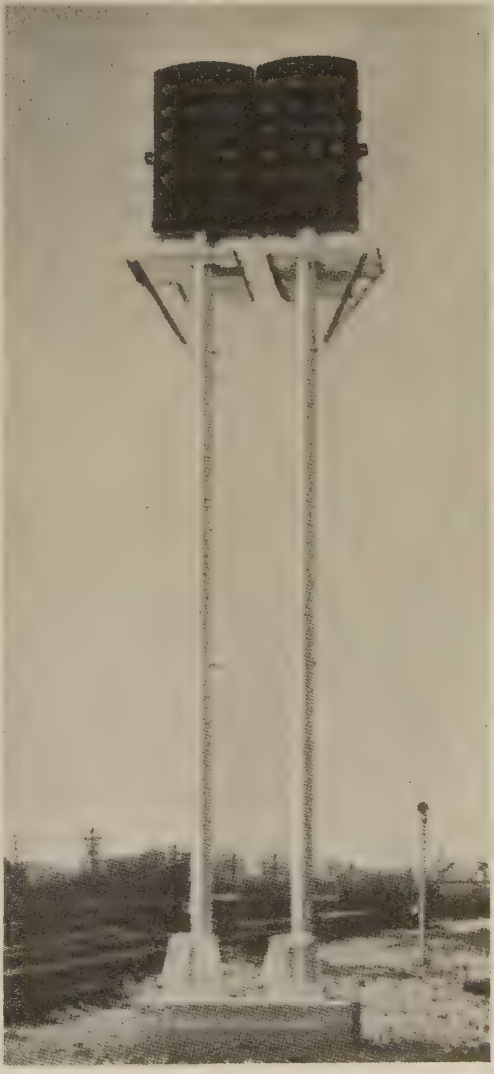
Keeps trains moving.

Between Florence yard and Pee Dee there are scheduled daily 20 passenger trains and about 20 to 24 freight trains, totaling 40 to 44 or more trains daily. Six of these trains go to or come from the Wilmington line.

Previously when passenger trains were due, southbound freight trains would be delayed at Pee Dee, or northbound trains, ready to depart from the yard at Florence, would be delayed. Now, with the entire area controlled from one machine, train movements are coordinated. The additional controlled crossovers at Mars Bluff, and the either-direction signalling on both main tracks, are used to advantage.

For example, a southbound freight train can be run from Pee Dee to Florence yard on track No. 2 while a southbound passenger train is run around it on track No. 1 between Pee Dee and Florence. Also, if a northbound freight is ready to leave the yard, it may be run out on track No. 2 as far as Mars Bluff, or to « WN », while, in the meantime, a passenger train of either direction can be routed over the other track. This flexibility has proved an important aid in preventing train delays, and in getting trains out of the yard with a minimum of delay.

A new one-story brick and concrete tower was constructed at the yard office to house the new five-foot Style « C » C. T. C. control machine, other C. T. C. equipment and batteries. This tower, designated



Yard track indicator keeps freight trains moving.

« WG », is on the west side of and at about the midway point of yard. The panel of the control machine has 9 levers for the control of 12 switches and cross-overs, 11 levers for control of 22 searchlight signals, and 3 push-to-turn levers for control of 5 electric locks on hand-throw switches.

Yard track indicators.

All southbound freight trains enter Florence yard at the north end. A part of the yardmaster's job is to be sure that a yard track is clear for each approaching train to enter. Previously, when approaching the yard, each southbound freight would stop to allow the head brakeman to telephone to the yardmaster to find out what yard track was available. This caused considerable delay. Therefore, as part of the new yard project, a lamp-type yard track indicator was installed at the north end of the yard.

This indicator has two panels, each with four vertical rows of five lamps each. The display of a track number in the indicator is controlled by the yardmaster, the controls being transmitted from the yardmaster's office to the indicator location — a distance of 6 600 ft. — on two two-wire line circuits, using code control system.

The panels for controlling the yard track indicator are mounted on a small table at the left of the yardmaster's desk. Each of the two panels at the top has ten telephone-type keys which the yardmaster operates to send out line codes to the yard track indicator. When the train arrives and enters the yard, the display of the track number of the indicator is canceled by sending out a cancel code.

This C. T. C. project was planned and installed by A. C. L. forces, under the direction of J. S. Webb, chief engineer communication and signalling; the principal items of signalling equipment and the yard track indicator equipment were furnished by the Union Switch and Signal Division of the Westinghouse Air Brake Company.

NEW BOOKS AND PUBLICATIONS.

[**656** .28 (42)]

Lt. Col. G.R.S. WILSON, Chief Inspecting Officer of Railways, Ministry of Transport (Great Britain). — **Report to the Minister of Transport upon the Accidents which occurred on the Railways of Great Britain during the year 1951.** — A brochure (6 × 9 1/2 in.) of 68 pages, with graphs and numerous tables. — 1952, Her Majesty's Stationery Office, York House, Kingsway, London, W.C. 2 and 423, Oxford Street, London, W. 1. (Price : 2 shillings net.)

This report is a substantial analysis of the accidents which occurred on the British Railways during 1951.

It gives with appropriate commentaries the statistics comparing the figures for the year 1951 with those of 1950 and the average of the years 1946 to 1950. Certain statistics however go back even further.

The accidents are grouped under three headings :

Class I. — Train accidents:

Class II. — Traffic accidents, and

Class III. — Accidents apart from the traffic.

In addition, in each of these three classes, the accidents affecting persons are subdivided as follows :

a) passengers; *b*) railway employees, and *c*) other persons.

After a general examination, the report studies the situation revealed by these figures and by the inquiries and reports to which they gave rise. A classification of the causes has been made and the author gives the proceedings and measures taken.

In the chapter dealing with class I (train accidents), there is a summary of the reports dealing with those which gave rise to a Departmental inquiry. These reports are published and contain recommendations on preventative measures. The publicity thus given has, in the past, had a very good effect in improving the safety.

In the conclusion, the author returns to the fact already pointed out, namely that the year 1951 was not so good from the point of view of train accidents as the two preceding years, which had shown remarkable results. This slight decline comes after an improvement, which has been steadily maintained throughout the years 1948, 1949 and 1950, and must be attributed to various unfavourable factors. The latter have been overcome and better indications recorded.

The appendices give more detailed statistics. In particular, there is an analysis of the primary causes of train accidents during 1950 and 1951 and a classification of accidents to trains and damage to rolling stock and the permanent way by kind and by Regions.

Regarding these latter, their number has decreased compared with the previous year, and this is due to the fewer broken couplings, the most important cause. There were however no collisions, nor derailments due to broken couplings on passenger trains. The general use of the continuous brake on goods wagons is now under consideration.

Special attention is given to accidents affecting railway employees of the traffic and other departments. Here the figures for 1951 are encouraging. The author sees in them a stimulant towards new progress, for which he praises the management, the unions and the men themselves.

E. M.

[**385** (05 (43)]

Archiv für Eisenbahntechnik *(Railway technical archives)*. Supplement to the monthly review *Eisenbahntechnische Rundschau, E.T.R.*, first issue. — One brochure (8 1/4 × 11 13/16 in.) of 49 pages, one plate and numerous figures. — 1952, Darmstadt, Carl Röhrig-Verlag O.H.G., 8, Stephanstrasse. (Price : 8.50 D.M.; for subscribers to the *E.T.R.* review 7 D.M.)

The new review ETR, published by Carl Röhrig-Verlag, at Darmstadt, the first number of which we reviewed in our *Bulletin* for August 1952, proposed to publish reports on various technical questions arising in the railway field. It was the intention of the editors and publisher to issue a quarterly supplement reserved for articles of some particular technical scope. This issue is the first of this series.

It contains four notes, the subjects of which we are going to describe.

In the first, Dr. Eng. P. GOTTSCHALK, studies the possibilities of the outlet from the Hohenzollern Bridge, at Cologne, rebuilt as a double track bridge, whereas the old bridge had four tracks. He shows how, thanks to the progress made in operating methods and the development of signalling almost the same number of trains can cross this bridge as used the old bridge in 1939.

Dr. C. POPP, Engineer, gives a study of the effects of shocks on the superstructure of railway lines due to flats on the wheels. He shows that this alteration in the shape caused by sliding of the wheels can be the cause of a considerable increase in the stresses on the rails. Numerical tables give the results of researches made on flats of various sizes, various materials and over a wide range of speeds.

As a complement to the preceding study, the Engineer H. RUBIN gives an account of the measurements he has made of tensions in the rails by means of up to date equipment, in collaboration with the Research Department of the German Federal Railways. The values found in practice are in complete agreement with the theoretical conclusions of Dr. Eng. POPP.

The fourth note, by Professor Friedrich HARTMANN deals with « the principles of the construction and new designs of points and connecting lines ». A specialist in this subject, the author develops the points of view which make it possible to improve the behaviour and life of points, allowing of high speeds whilst reducing the lateral shocks when the points are run through onto the branch line. He describes, with the necessary drawings, the new types of points laid down in the plans for laying track, especially branch lines and junction-crossings laid on curves.

E. M.

GERMAN TRANSPORT EXHIBITION
(Munich, 1953).

The great German exhibition of Transport and Communications is to be held in Munich, from the 20th June to 11th October 1953 (D. V. A., 1953).

For the first time since the war, this exhibition will cover the entire aspect of present day transport and communications as well as their linked up problems. This exhibition will go back in tradition to that of 1925, also held in Munich, which had more than three million visitors.

The city of Munich is making great preparations to live up to its reputation for organising exhibitions and fairs.

The park in which this huge exhibition is to be held (Münchener Ausstellungspark) has been completely modernised. New pavilions are to be erected as well as a new Congress Hall.

The area available for exhibitors and visitors is 50 000 m² (60 000 sq. yards) in the pavilions and 140 000 m² (166 000 sq. yards) in the open, a total of 560 000 m² (670 000 sq. yards). The installations will include amongst others a large station, a telpher railway line, a Lilliput railway, an aerodrome for helicopters and a Luna-Park. A certain part of the Munich-Riem airfield will be linked up with the exhibition on the Theresienhöhe.

The exhibition will cover all sections of modern traffic, railway services, road services, maritime services, air services, postal services and telegraph and telephone services, the consignment and storing of goods, as well as tourist services.

In addition, an exhibition organised by the industrial firms interested in the production of materials used for transport and its accessory equipment will be open to visitors.

(Communicated by the Press Department of the « Deutsche Verkehrsausstellung », Munich 1953.)

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